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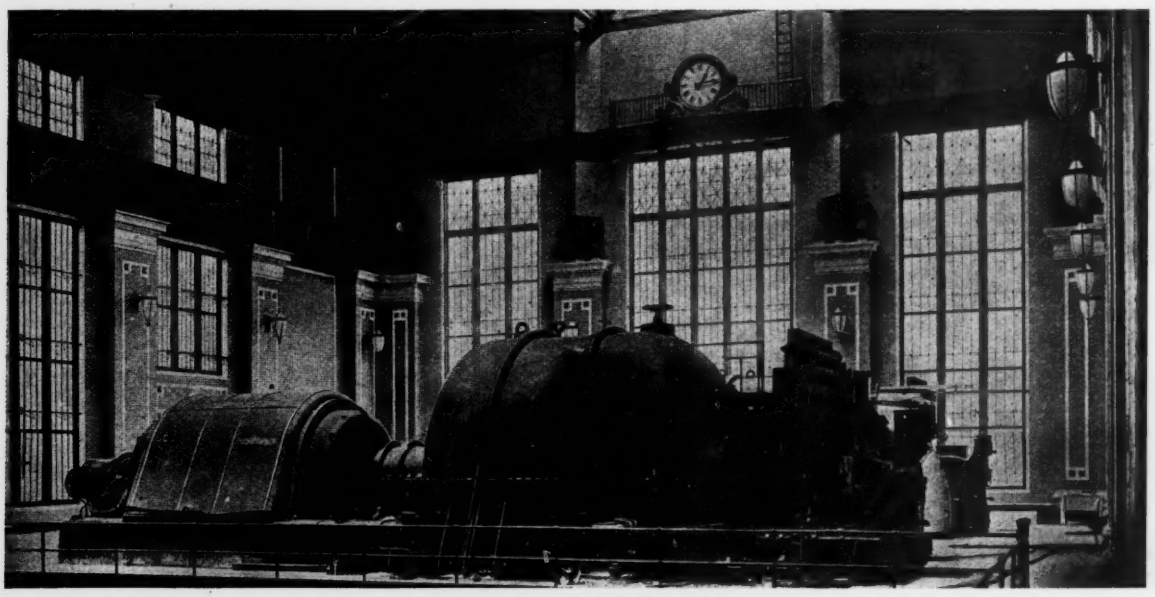

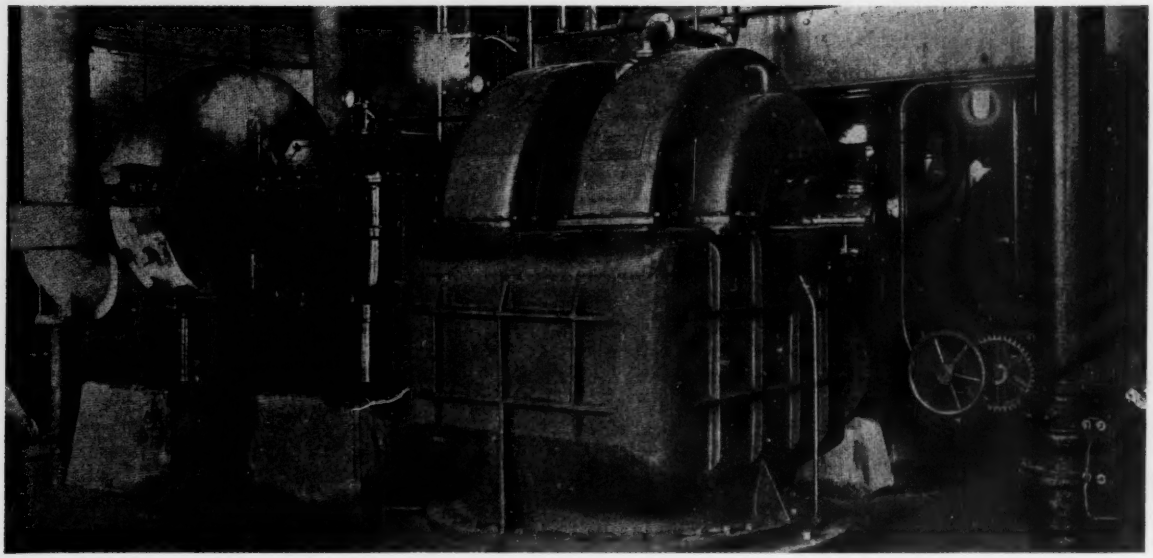

STEAM TURBINE DRIVE FOR MAIN UNITS AND AUXILIARIES IN MODERN CENTRAL STATION

Steam Turbines

TYPES AND CLASSIFICATION

THE SIMPLE STEAM TURBINE which in many respects resembles the water turbine consists essentially of a wheel or cylinder carrying on its periphery, uniformly distributed, a series of curved blades against which a jet of steam issuing from nozzles or guide passages is caused to impinge and thereby produce rotation. To obtain most satisfactory results the impinging jet must be formed in a nozzle of such shape as to provide the highest velocity obtainable and the wheel blades must be of such shape and arranged to run at such speed as to abstract the largest possible amount of the velocity energy from the jet before leaving contact with it.

Although, generally speaking, there are but two types of steam turbines, the reaction or pressure type, and the impulse or velocity type, developments brought forth characteristics and features necessitating further classification as indicated in Fig. 1.

Underlying principles of the reaction and impulse types may be understood best by means of simple prac-

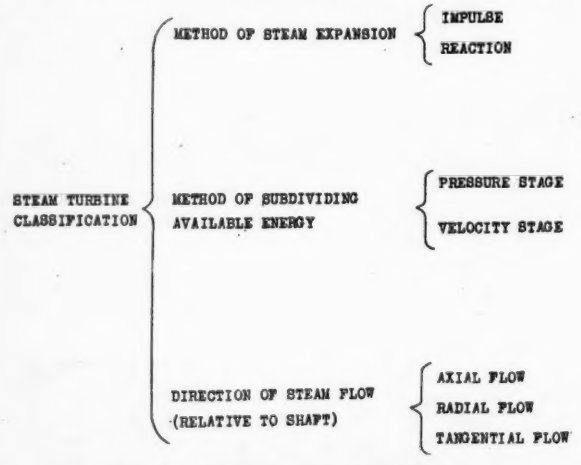

FIG. 1. TYPES OF STEAM TURBINES

tical illustrations. Shown at A in Fig 2 is a stationary nozzle, N, from which is discharged a jet of steam against plate P, and although both of these members are fixed there exists a tendency to force them apart. If, therefore, instead of having plate P fixed it is suspended from above by means of a hinge joint and is free to swing, the impulse of the jet against it causes the plate to be pushed from the nozzle. Or, if as at C, plate P is stationary, but nozzle N is free to swing on trunnion T, the reaction of the jet within the nozzle forces it backward.

Curving of the plate as at D prevents to some extent the breaking up of the jet and the resulting loss of energy although the pressure against the plate in this instance is the same as at A, and is caused wholly by the impulse of the jet. With further curvature of the plate, as at E, the steam is directed backward toward the nozzle and as a result, due to the impulse of the jet as it impinges on the plate and the following reaction as it leaves the plate, neglecting frictional losses, the available pressure against the plate is twice that realized with the scheme indicated at D.

Assuming V feet per second as the velocity of impact of the jet, and in view of the fact that after striking the plate this velocity is reduced to zero feet per second, it is evident that the force exerted against the plate is the same as that originally required to impart a velocity of V feet in one second to the mass of steam in the jet. This represents an acceleration of V feet per second and as force is equal to the product of mass times acceleration, the force acting against the plate will be equal to M times V, where M is the mass of steam in the jet, and V the acceleration, which in this case equals the velocity.

Then, with the plate moving in a like direction as the jet, but with lesser velocity of V_1 feet per second the relative velocity of the plate with reference to the

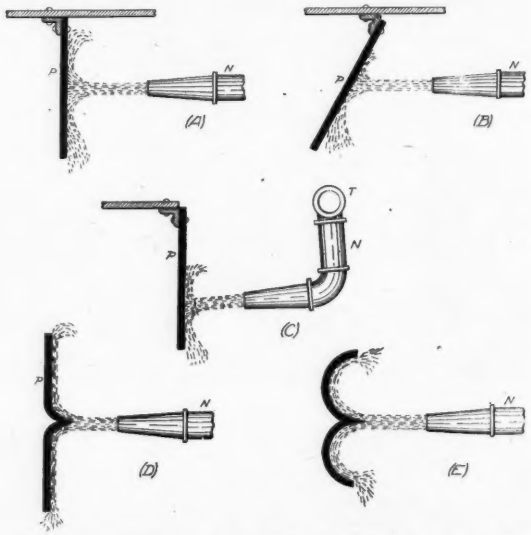

FIG. 2. JET ACTIONS AS FOUND EMPLOYED IN THE STEAM TURBINE

jet will be V minus V_1 and the actual corresponding force against the plate will be equal to M $(V - V_1)$.

Work being equal to the product of force times the distance through which it acts, and as force F in one second acts through a distance of V_1 feet, the work done on W is equal to FV_1 or M $(V - V_1)$ V_1 foot pounds.

But with the plate curved as at E, and with the direction of the jet almost completely reversed, the force as stated above acting against the plate is twice that realized at D, so that neglecting losses due to friction, the total force against the plate is equal to 2F and as a consequence the work done is also twice that with the divided plate or W is equal to $2FV_1$, or, 2 M $(V - V_1)$ V_1 foot pounds.

It is for this reason that turbine vanes are made so as to reverse the direction of the jet as much as possible, although complete reversal is not practicable due to the

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necessity of allowing some clearance for the escape of the deflected jet.

Types of Turbines

In practice, impulse turbines operate with some reaction, and reaction turbines with some impulse, but each type is designated according to whether its predominating feature is impulse or reaction.

In Figs. 3 and 4 are shown partially in section elementary forms of impulse wheels. With the shape of blade indicated in Fig. 3, the wheel is caused to rotate solely by the impulse of the jet of steam against the blade and as with the wheel held stationary the steam will issue from between the blades in a direction parallel to the shaft, this wheel may be considered as of the pure impulse type. Usually, however, the blades are curved as in Fig. 4, and as a result, although termed impulse, this wheel acts by both impulse and reaction.

With these arrangements and the form of nozzle employed the steam is expanded within the nozzle, wherein it attains a high velocity and in passing through the wheel gives up a part of its kinetic energy to the revolving blades and leaves the wheel at a lower velocity but at the same pressure at which it left the nozzle.

Shown in Fig. 5 is the simplest form of reaction turbine, the underlying principle of which is practically identical to the scheme indicated at C, Fig. 2. Steam enters trunnion T and flows radially outward through hollow arms AA until it discharges through nozzles NN. In the more practical form of this type of wheel illustrated in Fig. 6, the steam, after being but partially expanded in the nozzles impinges upon the blades and in passing between them acquires additional velocity by further expansion, thereby causing rotation of the wheel by both impulse and reaction, although due to the continued expansion of the steam within the wheel, this form is generally referred to as of the reaction type. Generally no nozzles are employed, the steam being admitted through guide passages provided for the purpose and placed around part or the entire periphery of the rotor. The pressure of the steam between the blades is greater than that of the steam beyond the wheel.

In the single-wheel impulse type of turbine the jet acquires velocity of exceedingly high values and as with the jet impinging upon the blades in a direction parallel to their travel an efficiency of 100 per cent may be realized with a blade of velocity of approximately one-half that of the jet it is evident that the rotors of this type of turbine must operate at a high rate of speed.

This arrangement, however, in addition to being objectionable for mechanical reasons, results in considerable loss due to the relative high velocity of the exhaust steam and to the friction of the bearings brought about by the high speed of rotation. As a consequence, single-wheel turbines are built only in the smaller sizes and then generally only when equipped with reduction gears so as to provide a speed more suitable to the requirements of the driven machine.

In large machines, however, the elimination of such high speeds and reduction gears is not only desirable but necessary and to adapt this type of machine for service requiring in excess of 500 hp., compounding is resorted to. Ordinarily this consists in the employment of two or more wheels mounted on a common shaft with guide vanes between the buckets, arranged as in Figs. 8 and 9. That in Fig. 8 is of the Riedler-Stumpf type, in which the steam flows through expansion nozzle N where its pressure is reduced to that of the steam surrounding the wheel, as indicated by the accompanying pressure diagram. After passing between the blades of the wheel wherein its velocity is reduced, it continues between guide vanes G, which reverse its direction of flow preparatory to passing through wheel W_2. At this point its velocity is again reduced.

In the Curtis type of compound impulse turbine, the steam as before passes through diverging nozzles NN, in which, due to expansion, the pressure is partially reduced while the velocity of the steam is increased. Wheel and guide passages, however, instead of being

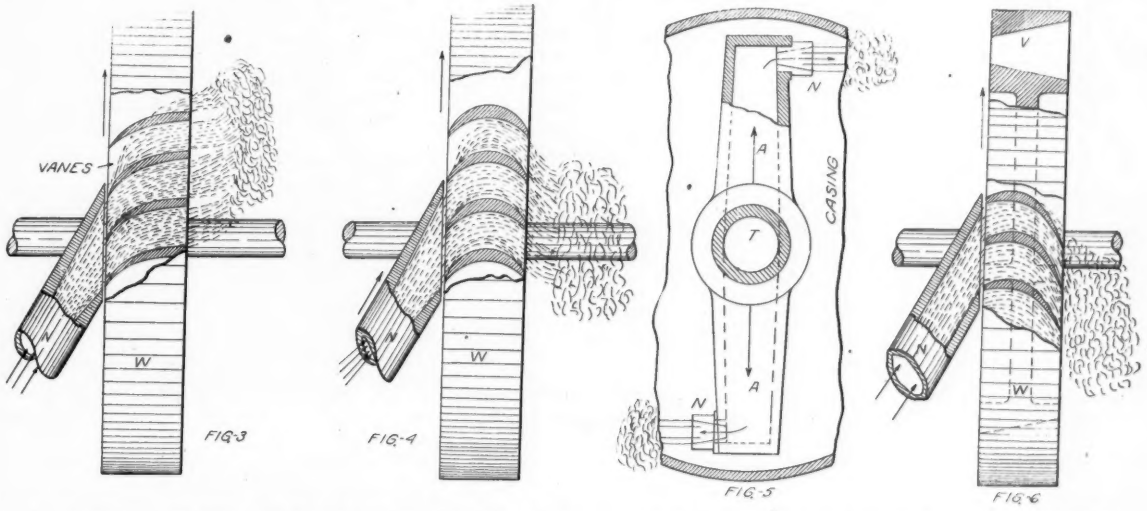


FIG. 3. ELEMENTARY IMPULSE WHEEL IN WHICH THERE IS PRACTICALLY NO REACTION

FIG. 4. A USUAL TYPE OF IMPULSE WHEEL

FIG. 5. SIMPLE FORM OF REACTION TURBINE

FIG. 6. A MORE PRACTICAL FORM OF REACTION TURBINE

of the same width as the Riedler-Stumpf turbine, diverge slightly from the point of entrance to the first wheel to the exit at the last, thus producing velocity and pressure diagrams of the form shown.

Compounding may be employed also in the reaction type of turbine as indicated in Fig. 10. As may be seen by the accompanying diagrams, the drop of pressure is gradual throughout the machine while the velocity increases as the steam passes through the guide vanes and decreases as it passes through the moving vanes.

Staging

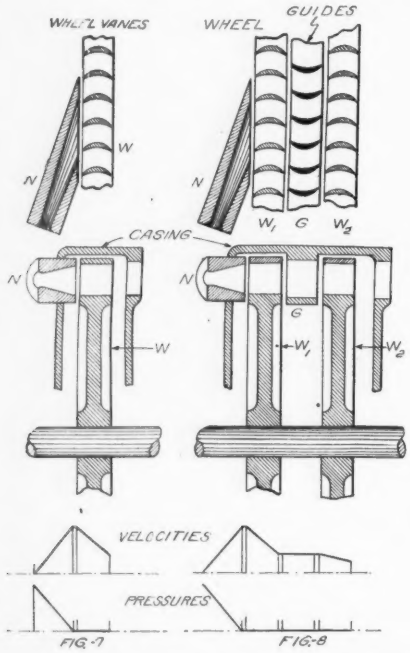
More gradual reduction in velocity and pressure than is realized through the ordinary compounding arrangements is obtained by the placing of two or more groups of wheels and guides in separate compartments as illustrated in Fig. 11, which shows the general scheme employed in the Rateau turbine. This consists essentially of a series of simple turbines like that pictured in Fig. 7, each of which is in a separate compartment; the velocity is reduced but once in each pressure stage and as the pressure is reduced as many times as there are compartments or stages, this is a multi-pressure, single-velocity stage turbine. The number of compartments which may be employed is limited only by mechanical design details, but in every case there should be at least sufficient so that the steam need not drop more than 0.4 of its initial pressure when flowing from one compartment to the next. Diverging nozzles are not required, but to accommodate the increasing volume of steam the areas of the guide passages are increased with each succeeding set.

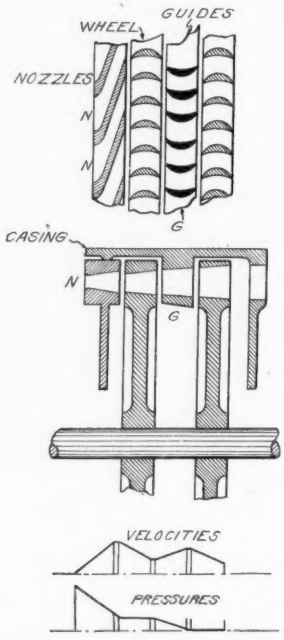
Where two or more pressure stages and two or more velocity stages exist, the term multi-pressure, multi-velocity stage is used and where two or more velocity stages and but one pressure stage, or one velocity stage and two or more pressure stages are provided, the terms multi-velocity, single-pressure stage, and single-velocity, multi-pressure stage, respectively, are employed; machines such as the Rateau, which are of the multi-pressure, single velocity stage type, are also referred to as multi-cellular.

Stage Arrangement

In impulse turbines the number of stages employed can be varied to a great extent with the pressure drop so subdivided that each stage has available a like number of heat units. As a result, all wheels may be of the same pitch diameter and have the same number of velocity stages.

Generally, however, in order to reduce the pressure the internal construction of these turbines is such as to cause the first stage to assume more than an equal share of the pressure drop, thereby lowering the maximum pressure to which the casing is subjected, simplifying the design of packing employed to prevent the escape of steam from the first stage to the atmosphere and reducing the rotational losses of the first stage wheel which generally are the greatest. In this case, the first stage wheel usually has 10 or more velocity stages, while the following wheels have but one velocity stage and greater pitch diameters, so as to increase their nozzle and bucket efficiency. In the first stage the nozzles usually occupy only a small portion of the pitch circumference, but with the decreasing pressure and increasing volume of steam the arc of admission increases from stage to stage until giving full peripheral admission in the last stages.





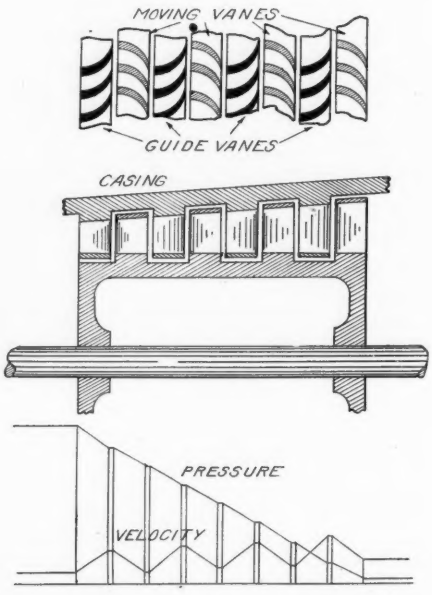


FIG. 7. SIMPLE IMPULSE WHEEL OF THE DE LAVAL TYPE
FIG. 8. COMPOUND IMPULSE TURBINE OF RIEDLER-STUMPF TYPE
FIG. 9. COMPOUND IMPULSE TURBINE OF CURTIS TYPE
FIG. 10. COMPOUND REACTION TURBINE—PARSONS TYPE

The number of stages in the reaction turbine is greater than the number employed in the impulse machine, steam and load conditions being similar. These stages are generally subdivided into so-called drums, which, on account of the decrease in pressure and increase in volume of the steam as it passes through the turbine, have increasing pitch diameters. Three drums provide a convenient arrangement in most reaction turbines and for the usual ranges of steam pressure encountered; the pitch diameters of the drums are in the ratio of 1 to $\sqrt{2}$ to 2, and with an energy distribution of approximately 1 to 1 to 1.5.

Characteristics

Some of the characteristics of the impulse or velocity type of turbine may be enumerated as follows: Few stages; expansion takes place in nozzles; large drop in pressure per stage; initial steam velocities 1000 to 4000 ft. per second; best efficiency obtainable when

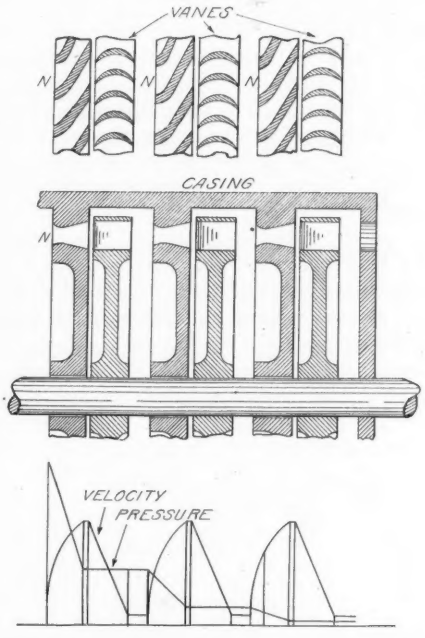


FIG. 11. COMPOUND IMPULSE TURBINE—MULTI-CELLULAR OR RATEAU TYPE

blade velocity is approximately half initial velocity of steam.

Characteristics of the reaction type of turbine are: Many stages; no nozzles; small pressure drop in each stage; steam velocities low, 300 to 600 ft. a second; blade velocities 150 to 400 ft. a second; best efficiency obtainable when blade velocity is nearly equal to highest steam velocity.

Impulse-Reaction Turbines

In the original Parsons turbine the reaction principle was used exclusively, but on account of the high velocity of rotor required to realize any satisfactory degree of efficiency a modification of this machine involving both impulse and reaction has been developed and is now used in some of the most prominent makes of turbines. The buckets employed are of such form so as to provide a curve at the entrance similar to that found in the impulse wheel and with a long, straight portion at the exit end forming a nozzle for the re-expansion of the steam and the creation of a reactive effect.

The gradually increasing sectional area of the steam passage between rotor and casing conforms roughly to that of an expanding nozzle and as a result the steam as it passes through the turbine experiences a gradual lowering of its pressure. But on account of this pressure drop, steam must be admitted around the entire rotor circumference, necessitating the use of comparatively short high-pressure stage blades and exceedingly close clearance between the ends of these blades and the interior walls of the casing. Due, however, to mechanical reasons, the minimum clearance which may be used allows the leakage of a comparatively large volume of steam, thus making the high pressure end of this type of turbine its least efficient portion.

On the other hand, the pure impulse type of turbine, in which the steam can generally be admitted through but a portion of the casing circumference, and where the pressure between the blades is the same as that surrounding the wheel, is more efficient in the high pressure stages than the reaction turbine, but on account of the presence of moisture in the steam it is less efficient in

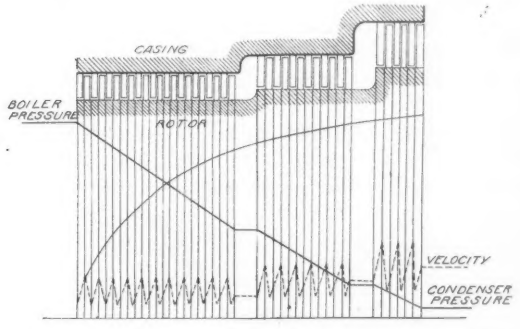


FIG. 12. SCHEME OF IMPULSE REACTION TURBINE

the later stages. If, therefore, as is done in some turbines, such as the Westinghouse-Parsons machine, the high pressure stages are equipped with impulse wheels and the intermediate and low-pressure stages with reaction or combined impulse and reaction wheels, the advantages accruing from the use of both impulse and reaction wheels may be realized.

In addition to the higher degrees of efficiency resulting from such an arrangement, more reasonable speeds, decreased number of high pressure stages, and shorter rotors and casings, eliminating considerable trouble otherwise experienced on account of expansion, may be obtained.

Miscellaneous Types

As indicated in Fig. 1, steam turbines may be classified further according to the direction of steam flow, as axial, radial and tangential. Of these the former arrangement, that is, the axial, in which the path of steam flow lies parallel to the shaft, is the most widely employed, especially in machines of the larger sizes. And while a number of turbines employing the radial flow principle have been developed, inherent mechanical difficulties have limited their general adaptation.

In the Terry turbine the steam flows tangentially along the rotor. The wheel has buckets or semi-circular recesses milled into its periphery, mounted around which are groups of nozzles with contiguous reversing chambers. The steam is expanded down to the exhaust pressure in a single nozzle, and, after striking one side of the wheel bucket, is reversed in direction, leaving the opposite side of the same wheel bucket. It then enters a stationary bucket or reversing chamber, which re-directs the steam into another bucket of the same wheel. This operation is repeated as often as is necessary to reduce the kinetic energy in the discarded steam to a relatively low value.

Another turbine of the tangential flow type is the Sturtevant, in which the action of the steam is similar to that in the Terry.

Arrangement of Turbines

REASONS FOR VERTICAL, HORIZONTAL, SINGLE AND DOUBLE-FLOW, BLEEDER, MIXED-PRESSURE, EXHAUST AND REVERSING TURBINES

IN THE development of steam turbines, one of the great difficulties encountered was to construct a machine so perfectly balanced that no vibration would occur at the extremely high speeds encountered. One of the early inventors, De Laval, solved the problem by making the balance as nearly accurate as possible, then depending upon the shaft to deflect sufficiently to allow the turbine to turn upon its true axis through the center of gravity. Such a solution depended upon a shaft of comparatively great flexibility, so flexible, in fact, that it would not long endure the lateral strains encountered when large units were placed with the shaft in a horizontal position. Among the early turbines, which ran at extremely high speeds, the vertical shaft was used almost exclusively and even today, among the medium size turbines running at moderate speeds, the vertical shaft arrangement is common practice; in large units, however, the vertical type is disappearing. The advantages of the vertical type were well stated by W. L. R. Emmet before the American Society of Mechanical Engineers as follows:

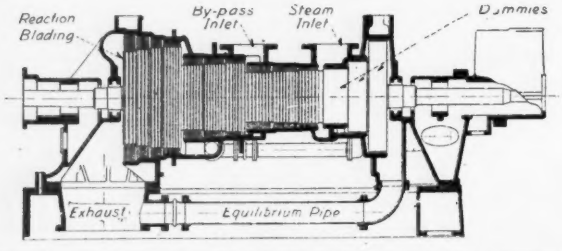


FIG. 1. WESTINGHOUSE-PARSONS STEAM TURBINE ILLUSTRATING SINGLE FLOW TYPE AND METHOD OF OVERCOMING END THRUST

"The relative positions of the revolving and stationary parts are definitely fixed by the step bearing. The stationary part is symmetrical, easily machined and free from distortion by heat. The shaft bearings are relieved from all strain, and friction is practically eliminated. The shaft is free from deflection and can be made of any size without reference to bearings, which can be placed where convenient and operated with surface speeds which would not be practicable with the horizontal arrangement.

"These features make possible the use of a very short shaft and consequently the longitudinal spacing of moving and stationary parts is very little affected by temperature differences. The turbine structure affords support and foundation for the generator. The cost of foundations is very small, and the solidity and alinement of foundation, is not of vital importance. Much floor space is saved. All parts of the machine are conveniently accessible. Failure of lubrication cannot injure the shaft or other expensive parts."

As the science of turbine design advanced, speeds have been reduced, methods of balancing have improved, and as the units have become larger, the horizontal type, particularly in large sizes, has been most commonly installed. Among the advantages claimed for the horizontal machine are: No step bearing which must be operated with the greatest of care and attention under high pressure oil or water. This arrangement permits the construction of the largest units and accommodates more stages, thus permitting lower speeds. The various parts of turbine and generator are more readily accessible. Ventilating ducts can be more conveniently arranged. Steam and exhaust piping may all be put in basement or trenches.

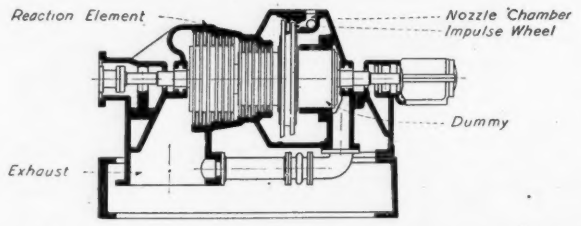


FIG. 2. SECTION THROUGH COMBINED IMPULSE AND REACTION SINGLE FLOW WESTINGHOUSE TURBINE

Among the problems met with horizontal turbines of the reaction type not encountered with the vertical arrangement, is the elimination of end thrust. In small sizes, this is taken care of by thrust bearings; in medium sizes, balancing pistons are used to some extent; while in large sizes, the double-flow arrangement is common practice. Although the last two methods will permit of perfect balance, there is still some oscillatory motion which must be eliminated so that a thrust bearing usually of the labyrinth type, is provided in most cases. This bearing permits of definite adjustment of the rotor position so that clearances between revolving and stationary blades are maintained accurate.

Figure 1 is a good illustration of the balancing piston method of overcoming end thrust. Here it will be noted that for each section of the rotor carrying blades there is a blank section of the same diameter at the other end;

the steam pressure against balancing sections is equalized by passages from one end of the rotor to the other, resulting in a perfect pressure balance.

In the combination impulse and reaction, single-flow type, illustrated in Fig. 2, the reaction blading of the first or high-pressure section has been replaced by an impulse element. Steam is admitted to the nozzle chamber, expanded in the nozzles and discharged against a portion of the periphery of the impulse wheel, thence to the medium and low-pressure sections which constitute the reaction element. The entering steam is confined in the nozzle chamber, and in passing through the nozzles and impulse wheel, its pressure and temperature are materially reduced. As the nozzle chamber is cast separately from the main cylinder, the temperature and pressure differences to which the cylinder are subjected are correspondingly lessened. A dummy element is employed in all single-flow turbines to maintain an axial balance and thus prevent end thrust due to the passage of steam through the reaction elements. These dummies are subjected to steam pressures or vacuum equivalent to those on the active elements of the drum, the connections being made through steam pipes or passages through the drum.

In the double-flow turbine of the combined impulse and reaction type, a section of which is shown in Fig. 3, the principal advantage is a gain in capacity, for the maximum economical capacity of a single-flow turbine is limited by the rotative speed which in turn is limited by the allowable stresses due to centrifugal force. The double-flow design is in effect two single-flow turbines made up in a single rotor in one casing with a common inlet and two exhausts. Steam enters the nozzle chamber, acts on the impulse element and then the current divides, half the steam going each way from the impulse element through the reaction sections. This arrangement automatically takes care of axial thrust.

The semi-double-flow turbine is a modification of the double-flow type in which the intermediate pressure reaction element on one end of the drum is eliminated, a dummy taking its place for axial balance. At the other end, the diameter of the intermediate element is made larger than in the double-flow type, thus accommodating the same amount of steam, permitting a shorter drum of greater diameter. This type has been developed to take

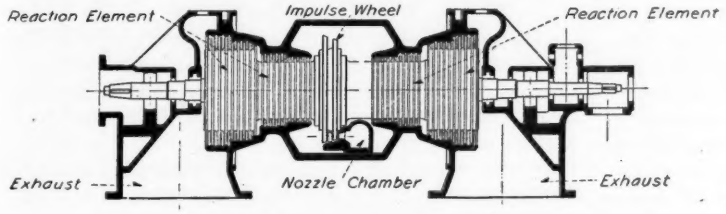


FIG. 3. SECTION OF DOUBLE FLOW COMBINED IMPULSE AND REACTION WESTINGHOUSE TURBINE

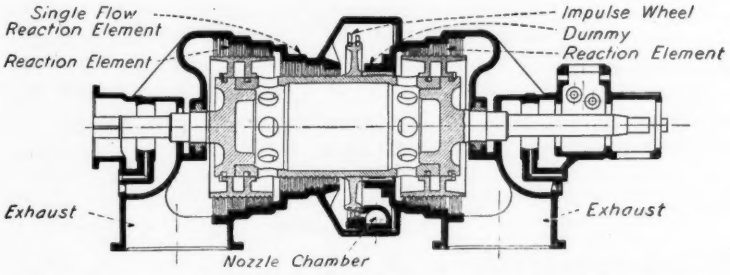


FIG. 4. SECTION OF WESTINGHOUSE SEMI-DOUBLE-FLOW TURBINE

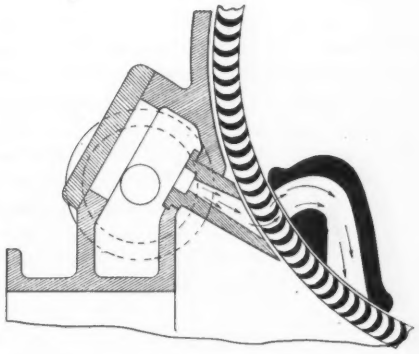


FIG. 7. NOZZLE AND REVERSING BUCKET OF TURBINE, ILLUSTRATING RADIAL FLOW TYPE

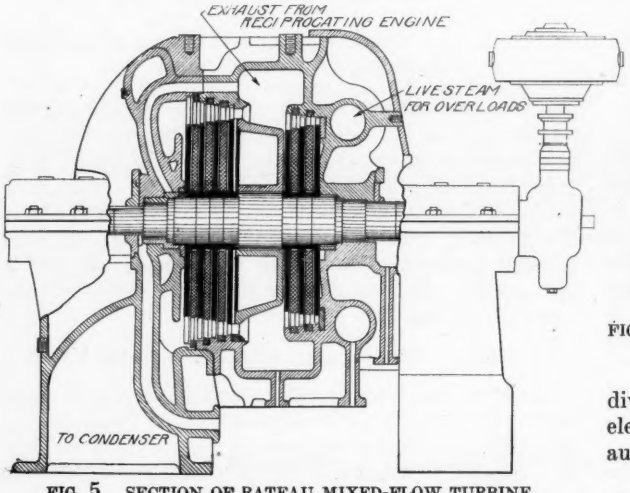


FIG. 5. SECTION OF RATEAU MIXED-FLOW TURBINE

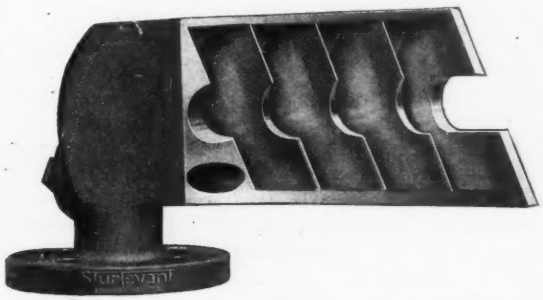


FIG. 6. NOZZLE AND REVERSING BUCKETS OF STURTEVANT TURBINE, ILLUSTRATING TANGENTIAL FLOW TYPE

a place, in capacity, between the single-flow and double-flow machines of the same speed.

What is termed a mixed flow turbine, a cross section of one type being illustrated in Fig. 5, consists of two sections of blading, one for high, the other for low-pressure steam, and is designed for special conditions sometimes met in industrial plants where the reciprocating engine is not in operation at all times, or the exhaust from it is not always sufficient, so that the supply of exhaust steam is not always available for the turbine. During these periods of insufficient or no exhaust steam, high-pressure steam is passed through the high-pressure element of the turbine, then passes to the low-pressure section combined with what exhaust steam may be coming from the reciprocating engine. The supply of live steam is controlled automatically by a governor which admits or cuts off the steam as conditions require.

Plants that are required to furnish steam for heating or process work at a pressure above that of the exhaust from the turbine are well served by what is known as a bleeder turbine when the requirements for low-pressure steam do not demand all the exhaust steam from the turbine. In this type, steam is taken from the stage furnishing the desired pressure, thus diverting the steam from the low-pressure stages and calling for more high-

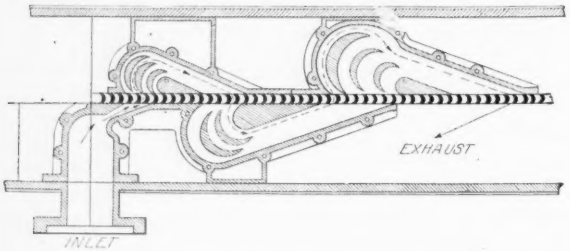


FIG. 8. PLAN FOR COMPOUNDING WITH ONE ROW OF MOVING BLADES

pressure steam to deliver the same power. This type is the reverse of the mixed-flow turbine, the principal difference in construction being in the relative sizes of the high and low-pressure elements.

One of the distinct advantages of the steam turbine is its ability to work to advantage with the extremely low pressures obtained with modern condensers. This characteristic has led to the development of the low-pressure or exhaust steam turbine, which differs but little in general construction from the high-pressure turbine, but has a field all to itself. The following classification of low-pressure turbine installations is given by Edwin D. Dreyfus and shows well the varied and unique applications of this type of turbine:

a—Addition to a well-arranged and symmetrical alternating or direct-current power-generating station, condensing or non-condensing.

b—Extensions of poorly designed stations with dissimilar units.

c—Increased capacity for mills driven mechanically by main engine.

d—Mills with widely distributed engines, demanding more power.

e—Conservation of waste in intermittently-operated and reversing engines.

f—Variable alternating-current electrical load for low-pressure turbine with uniform mechanical or direct-current load for engine and vice versa.

g—Utilization of gas engine waste heat.

The nature of both the application of and the demand for continuous power from the low-pressure turbine has, in the different cases above enumerated, evolved the following systems:

1—Turbine without governor.

2—Simple low-pressure turbine having governor control, with supplementary live steam admission valve.

3—Use of synchronous motor providing an electrical and mechanical tie with the main belt or rope wheel engine.

4—Governor operating valve bypassing steam to condenser.

5—Heat regenerators and accumulators.

6—Mixed-flow turbine with high-pressure element.

7—Heat storage systems.

In marine work where the propeller is attached directly to the turbine shaft, some means must be provided for reversing the propeller during maneuvering operations; for this purpose, reversing turbines have

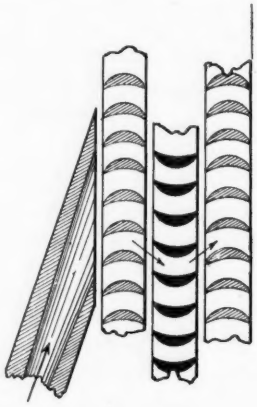


FIG. 9. USUAL METHOD OF COMPOUNDING AXIAL FLOW TURBINES OF LARGE CAPACITY

been designed. These consist essentially of two turbine rotors on one shaft with the blading arranged for opposite directions of rotation. The reversing element is of smaller capacity than that driving the ship forward, and is placed near its exhaust end. It is so arranged that the reversing element runs in a vacuum when the ship is going forward. The method of reversing the ship is very simple, consisting in turning the steam off one element and onto the other.

Placing of Nozzles, Stationary and Guide Vanes

Nozzles for steam turbines may be so placed as to direct the steam on the buckets at a slight angle, usually about 20 deg., to the tangent of the rotor in turbines of the radial or tangential flow types; while in turbines of the axial flow type, the nozzles are placed at the side of the rotating buckets directing the steam against them at an angle of about 20 deg. from the plane of rotation.

Nozzles for tangential flow turbines are usually combined with reversing buckets as shown in Fig. 6. With this arrangement, steam expanding through the nozzle impinges into one side of a rotor bucket, which changes

its direction of flow delivering it to the first reversing chamber which reverses the direction of steam flow, delivering it again to the rotor buckets. This process of reversing the steam flow continues until all the reversing chambers have been passed by the steam. Such nozzle segments are usually in one piece with the reversing chambers. The number of nozzles employed depends upon the capacity of the turbine and by a system of governing which puts into operation or cuts out the nozzles, the efficiency at various loads is maintained nearly constant.

Nozzles for radial flow turbines are arranged similarly to those in tangential flow types, but the nozzle is separate from the reversing chambers, which are placed on the opposite side of the rotor blades. Modern machines employing this arrangement have but one reversing chamber for each nozzle, thus passing the steam only twice through the rotor blades; this arrangement is shown in Fig. 7. For the same size rotor, this arrangement does not permit the use of so many nozzles as does the tangential flow type, as the space occupied by the reversing chamber is greater.

By far the most common arrangement of nozzles is for axial flow of the steam. With the nozzle by the side of the rotor blades, the steam strikes the blades and passes out at the other side. Velocity staging of this type is done in either of two ways; in the smaller capacities, the steam exhausted opposite the nozzle is caught in reversing blades and directed back upon the rotor blades, a process which may be continued three or four times as indicated in Fig. 8; in large turbines, stationary guide vanes placed opposite the nozzle direct the steam against a second row of revolving blades as shown in Fig. 9. This process takes place in impulse turbines through one or two sets of stationary blades for each pressure stage; in reaction turbines, the rotor blades form in reality expanding nozzles, the expansion taking place in small degrees all the way from inlet to exhaust, stationary guide vanes being placed between each two wheels change the direction of steam flow, directing it against the next set of rotor blades. In the impulse axial flow types, the steam nozzles may be evenly spaced all around the periphery of the rotor or only two or three may be employed. In the latter case, there is a section of stationary guide vanes opposite each nozzle, the portions between the sections being blank. This arrangement is employed in the Ridgway turbine, as illustrated on another page of this issue. As the pressure decreases through the turbine, the sections containing guide vanes lengthen until in the last sets they extend completely around the diaphragms. Where this arrangement is employed, the rotor diameter may be maintained constant without the blades becoming excessive in length.

Turbine Piping

PIPING SYSTEMS FOR VARIOUS TURBINE ARRANGEMENTS

WITH the high steam pressures and superheat carried in turbine plants, also the high steam velocities and high vacuums, piping becomes an extremely important feature of a steam turbine plant. Pipe sizes are figured from a basis of steam velocity. While for reciprocating engines, a steam velocity with saturated steam of 8000 to 10,000 ft. a minute is common practice, in turbine plants the velocity is increased to from 12,000 to 20,000 ft. a minute. There appears to be no standard rule for steam velocities, designing engineers depending largely upon their experience to guide them in this particular, but it is noted that the high velocities are employed where the pressure and superheat are also high. There is undoubtedly initial economy in using smaller pipes, particularly under high pressures, and with superheated steam the pipe friction loss is reduced. Another feature lending itself to higher steam velocities with turbines than with reciprocating engines is the continuous flow of steam.

In exhaust piping, according to Harding and Willard, the most satisfactory velocities range from 20,000 to 24,000 ft. a minute in condensing plants and 8000 to 10,000 in non-condensing plants.

Piping materials should be selected with great care, as it is well recognized that some metals, such as cast iron, used in valves and fittings, undergo changes in size when subjected continuously to the superheated steam temperatures commonly employed in modern turbine

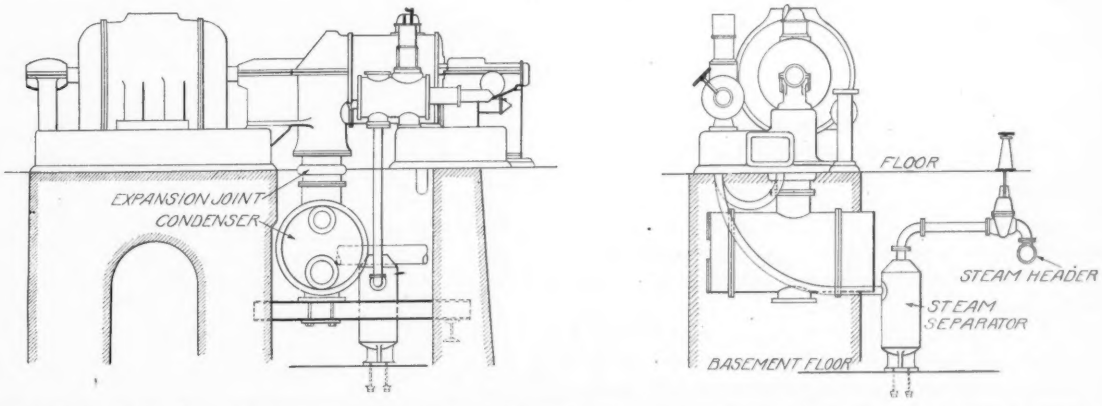


FIG. 1. SUGGESTED PIPING ARRANGEMENT WITH HEADER IN BASEMENT AND CONDENSER BENEATH TURBINE

plants. Satisfactory results have been obtained with wrought iron and steel piping and fittings, and cast steel valves. Pipe joints of the flanged or welded type give best results and many special forms have been devised. Those without gaskets are usually preferred, although more expensive. There are, however, certain types of flanges with the gaskets placed in grooves and others employing metallic gaskets that give good results under high-pressure and superheated steam conditions.

All connections should be made as short and direct as possible, allowing, of course, provision for expansion and contraction of the pipe line due to the change in temperature. Although the steam used for driving turbines is usually superheated and little condensation occurs in the piping, it is always wise to use a steam separator at or near to the turbine connection, for with the high velocities employed, slugs of water entering the turbine could do considerable damage to the blading.

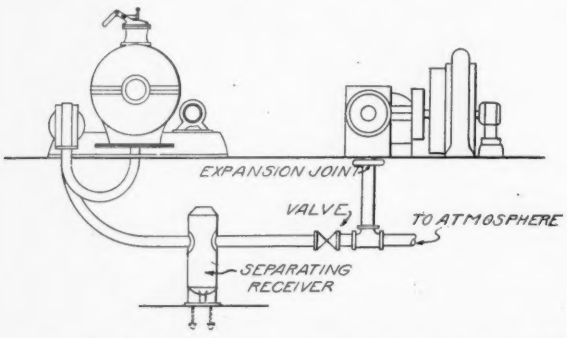


FIG. 3. PIPE CONNECTIONS BETWEEN ENGINE AND LOW-PRESSURE TURBINE

In Fig. 1 is shown a suggestion for connecting up a high-pressure condensing turbine where the steam header is in the basement and the surface condenser is located directly below the turbine. A stop valve is shown immediately at the header, so that the whole of the turbine piping may be cut off if necessary. The steam passes from the upper side of the header, where it is not likely to contain water, to a steam separator securely anchored to the cement foundation, thence to the turbine through long radius bends. The condenser is placed on I beams built into the turbine foundation and connected to the turbine nozzle by a copper expansion joint, the purpose of which is to prevent any undue strains being transmitted to the turbine from the expansion or contraction of the exhaust piping.

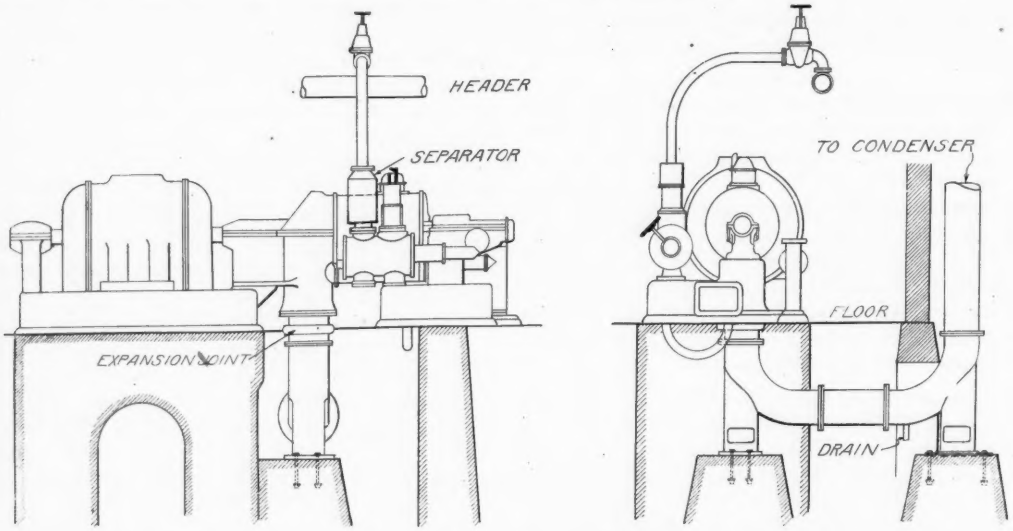


FIG. 2. PIPING SUGGESTION WITH HEADER IN ENGINE ROOM AND CONDENSER OUTSIDE

Where the steam header is located in the engine room and the condenser outside the building, the piping suggestion shown in Fig. 2 is a good arrangement. Here, as in Fig. 1, the steam is taken from the upper side of the header through an angle valve and passes through a long radius bend to a vertical steam separator attached directly to the turbine steam nozzle. Exhaust in this case passes through a foot elbow located directly under the turbine exhaust nozzle and securely anchored to the foundation.

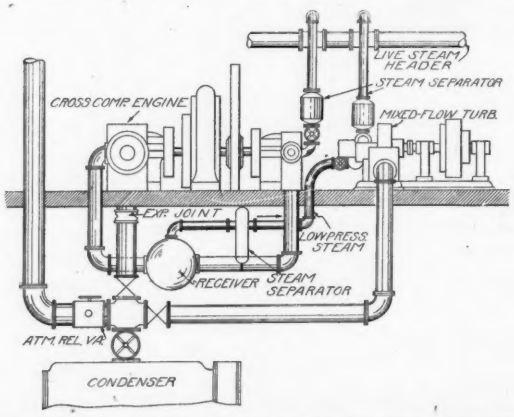


FIG. 4. PIPING ARRANGEMENT FOR COMPOUND CONDENSING ENGINE AND MIXED-FLOW TURBINE

Where a low-pressure turbine takes all or part of the steam exhausted from a reciprocating engine, an arrangement such as shown in Fig. 3 will give good results. It will be noted that the engine is provided with piping which permits it to exhaust through the separator to

the turbine, or to the condenser direct or to atmosphere. This allows for considerable flexibility in the use of the engine. While the turbine is taking the engine exhaust some provision should be made for relieving exhaust pressure on the engine in case the turbine does not require all the steam available. A back-pressure valve on the atmospheric exhaust line will serve this purpose.

In cases where the engine carries an intermittent load of considerable variation, if the exhaust is used to drive a turbine it is necessary to employ a regenerator to main-

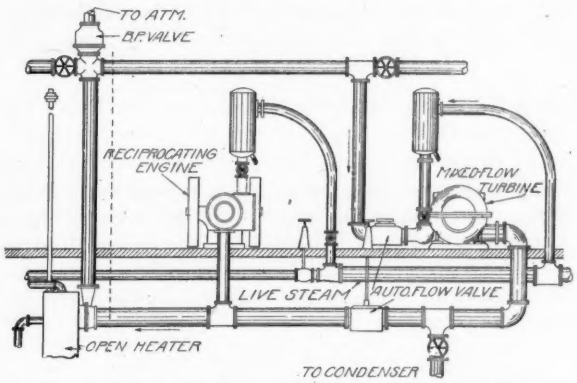


FIG. 5. MIXED-FLOW TURBINE ARRANGED TO FEED HEATING SYSTEM OR RUN CONDENSING

tain a constant supply pressure on the turbine. The piping for such an arrangement is similar to that shown in Fig. 3, except that the regenerator takes the place of the separator.

Occasionally in a plant containing a cross compound condensing engine, it becomes desirable to install a turbine to carry a part of the load separately from the main unit. The piping arrangement shown in Fig. 4 suggests

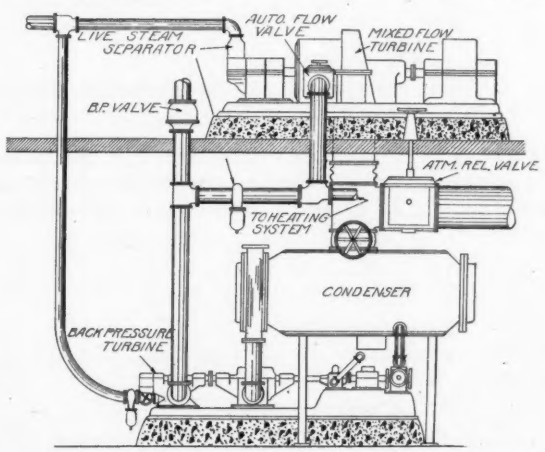


FIG. 6. BACK-PRESSURE TURBINE DRIVING AUXILIARIES, EXHAUSTING TO MAIN TURBINE OR HEATING SYSTEM

a solution to this problem. In this case, a mixed-flow turbine is used, taking steam either from the engine receiver or direct from the high-pressure header, and exhausting to the condenser that serves the engine.

In Fig. 5 is a suggestion for connecting up a mixed-flow turbine to increase the capacity of a plant and improve the economy of the existing reciprocating engine which exhausts to the heating system. With this arrangement the turbine may be operated on surplus exhaust from the engine, supplemented by live steam if necessary, and exhaust to the condenser, or when heating requirements are greater than can be supplied from the engine, the turbine takes high-pressure steam only and exhausts into the heating system.

For driving auxiliaries such as condenser pumps, boiler feed pumps, exciters and the like, turbines offer a flexible and economical driving element. Either a mixed-flow, back-pressure or low-pressure turbine may be used for auxiliaries. The mixed-flow machine offers several arrangements adapting it to service in almost any plant. It may take high-pressure steam exhausting to the intermediate pressure stage of the main turbine or to the heating system or to a condenser. When supplied with exhaust steam supplemented by high-pressure steam, the arrangement shown in Fig. 5 is frequently used.

Auxiliary turbines driven by high-pressure steam only and exhausting against a back pressure may be connected up to deliver low-pressure steam to the intermediate pressure stage of the main turbine or to the heating system as indicated in Fig. 6.

Low-pressure turbines used for driving auxiliaries must exhaust to a condenser; they may take steam from

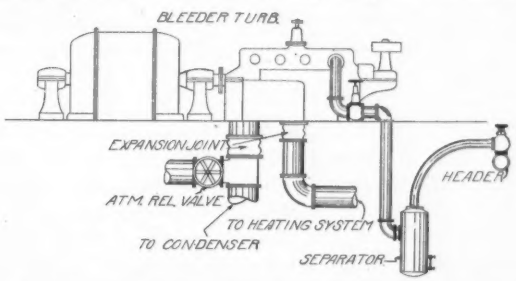


FIG. 7. PIPING ARRANGEMENT FOR A BLEEDER TURBINE SUPPLYING STEAM TO A HEATING SYSTEM

a bleeder connection on the main turbine, from the receiver between the high and low-pressure cylinders of a compound condensing engine or from the exhaust of a simple engine working against a back pressure slightly above atmospheric.

In industrial plants that require steam for heating purposes, the automatic bleeder turbine serves well the requirements. In these installations, steam is taken from an intermediate stage which supplies the heating system with steam at proper temperatures. With proper design the turbine will carry its power load steadily and give good regulation, at the same time furnish the steam at proper pressure automatically in varying quantities as desired. Figure 7 shows the piping arrangement to such a turbine.

One of the most recent developments in the arrangement of steam turbines is that employed by the Interborough Rapid Transit Co. of New York City, in a recent installation of two 60,000-kw. units. Briefly, this arrangement consists of three turbines installed side by side, the high-pressure unit being placed between the two low-pressure units. Each machine drives a separate generator, all three generators being connected together electrically. Governing devices are provided by which, in case of the failure of any unit, the others will continue to carry their share of the load.

Turbine Nozzles

DETERMINATION OF PROPER SIZES AND SHAPES. MATERIALS USED

NOZZLES add no energy to the steam, but are employed to give direction to the steam jets so that the greatest amount of energy may be utilized. Where the drop in pressure is to a low absolute value less than 0.58 of the absolute high pressure, a diverging nozzle, Fig. 1, is used, as this form is needed to prevent vibration and eddy currents. If the pressure beyond the nozzle is more than 0.58 of that in front of it, a nozzle with only converging outline is used.

Experiment has shown that steam expanding through any kind of orifice to a pressure less than 0.58, the original pressure will fall suddenly to 0.58, that initial pressure just beyond the orifice. There is a slight variation from this law, depending on the original pressure and the amount of moisture in the steam, but it is not sufficient to be of practical consequence.

Further drop in pressure takes place by a series of pressure oscillations, decreasing in amplitude until the pressure of the exhaust medium is reached. The diverging nozzle is designed to reduce these vibrations to a minimum, and direct the force of the jet upon the blades at the angle which experience has shown to be most effective.

Velocity of the jet depends on the number of heat units which are converted into velocity energy, and this, again upon the pressure drop. The energy of 1 lb. of steam at the initial pressure is 778 times the total heat at that pressure, and after expansion is 778 times the total heat at the lower pressure + ($V^2 \div 64.4$). If we denote initial conditions by the subscript 1, and final by the subscript 2, we have:

$$E_1 = 778\ H_1 \quad \ldots\ldots (1)$$

$$E_2 = 778\ H_2 + (V^2_2 \div 64.4). \quad \ldots\ldots (2)$$

As the energy is the same for both cases, $E_1 = E_2$ or $778\ H_1 = 778\ H_2 + (V_2^2 \div 64.4)$, where H denotes total heat as found from steam tables and V denotes velocity in feet per second.

WATER RATE FOR TURBINES. 100 LB. GAGE PRESSURE—ATMOSPHERIC EXHAUST

Rated Capacity Brake Hp.	Steam Consumption Lb. Per Brake Hp. Hr. Full Load
10	79
25	54
50	39
75	32.4
100	30.2
150	29.4
200	29.0
250	28.2
300	27.8
400	27.0
500	26.5
600	25.8
800	22.2
1000	22.0

Solving the last equation for V_2 we get:

$$V_2 = 224 \sqrt{H_1 - H_2} \quad \ldots\ldots (3)$$

This heat drop is reduced somewhat by the friction in the nozzle, Stodola giving this as 15 per cent and Roe as 10 per cent, the energy used in overcoming friction being reconverted into heat, which is reabsorbed by the steam. Actual final velocity V'_2 will then be:

$$V'_2 = 224 \sqrt{(H_1 - H_2)\ (1 - \%\ \text{loss})} \quad \ldots\ldots (4)$$

which for 15 per cent becomes $0.92\ V_2$, or for 10 per cent becomes $0.95\ V_2$.

In proportioning the nozzle, it is necessary to know the proper diameter at each point to pass the required amount of steam at the velocity due to the heat drop up to that point, and this depends on the weight of steam to be passed per second, W, the specific volume cubic feet per pound, v, and the velocity, V. Worked into a formula, it gives:

$$\text{Area in square inches} = Wv \times 144 \div V.$$

The two points of greatest importance are the throat and the exit. At the throat there has been no loss by friction, but if the steam was dry and saturated at the

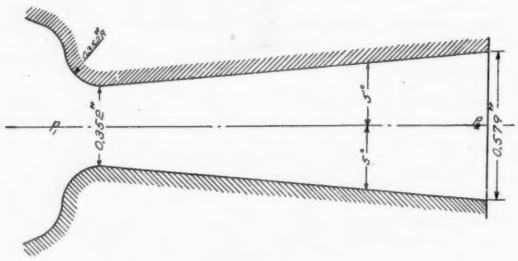


FIG. 1. FORM FOR NOZZLE FOR NON-CONDENSING IMPULSE TURBINE

start some of it will have condensed, and the volume per pound will be v'_t, less than v_t by the per cent of condensation. If we designate conditions at the throat by subscript t, we have for area in square inches:

$$A_t = \frac{Wv_t' \times 144}{V_t} \quad \ldots\ldots (5)$$

At the exit, the friction effect has entered, also further condensation has occurred, hence, for exit area in square inches:

$$A_2 = \frac{Wv'_2 \times 144}{V'_2} \quad \ldots\ldots (6)$$

The process is best made clear by an example. Required the form of nozzles for a 150 brake horsepower non-condensing turbine with five nozzles, to operate with dry saturated steam of 150 lb. gage pressure and 15.7 absolute back pressure.

Steam consumption can best be estimated from experience, and the table gives values as found for turbines from experiment. These are for 100 lb. gage initial pressure and atmospheric exhaust and must be corrected for other conditions as indicated.

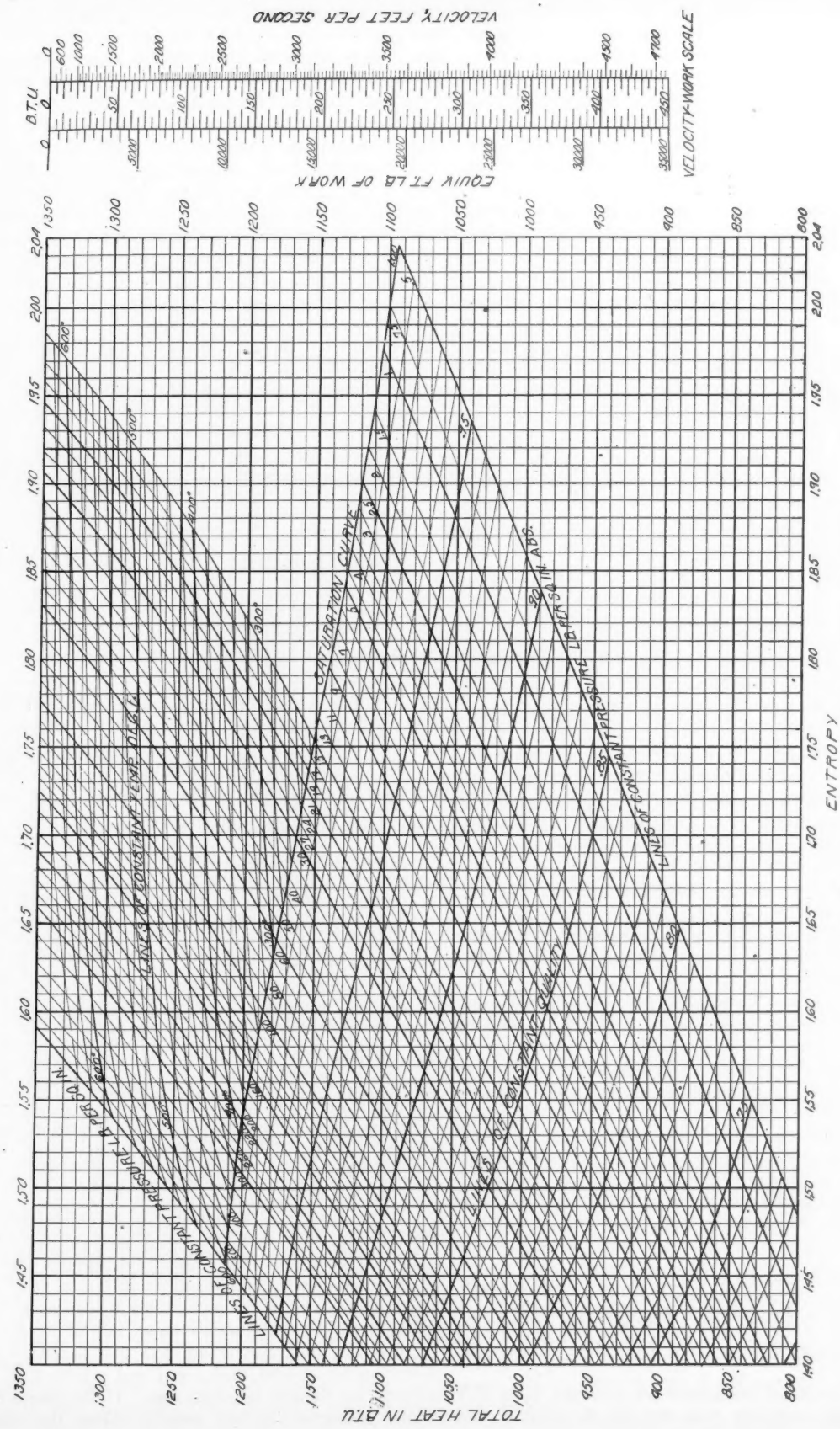

VELOCITY FEET PER SECOND
B.T.U.
EQUIV FT LB OF WORK
VELOCITY-WORM SCALE
TOTAL HEAT IN B.T.U.
ENTROPY
CALIBRATION CURVE
LINES OF CONSTANT PRESSURE IN LBS PER SQ IN ABS.
LINES OF CONSTANT QUALITY
LINES OF CONSTANT SUPERHEAT IN DEGS F.
LINES OF CONSTANT TEMPERATURE

FIG. 2. MOLLIER'S HEAT CHART FOR STEAM VALUES

For superheat, decrease the water rate 1 per cent for each 10 deg. F. superheat up to 100 deg. superheat and for each 12 deg. F. above 100 deg. superheat.

For higher pressures, decrease the water rate 2 per cent for every 10 lb. increase in pressure, and vice-versa.

For increase of back pressure, increase the water rate 3 per cent for 2 lb., 5 per cent for 3 lb., 7.5 per cent for 4 lb, and 10.5 per cent for 5 lb.

For condensing turbines, decrease the water rate from non-condensing values, 14 per cent for 18 in. vacuum; 16 per cent for 20 in.; 20 per cent for 22 in.; 24 per cent for 24 in.; 30 per cent for 26 in.; 34.5 per cent for 27 in.; 40 per cent for 28 in.; 44 per cent for 28.5 in.; 48 per cent for 29 in.

For 150 brake horsepower, the table shows a probable steam consumption of 29.4 lb. at 100 lb. pressure. As this turbine is to take steam at 150 lb. gage, the steam consumption would be reduced 10 per cent or to 26.5 lb., and as back pressure is about 1 lb. gage the increase would be about 1 per cent or to 26.76 lb. per hp. hr., or practically 27 lb.

As the turbine is to deliver 150 brake horsepower and has five nozzles, each nozzle must pass per second $150 \times 27 \div (3600 \times 5) = 0.225$ lb. of steam.

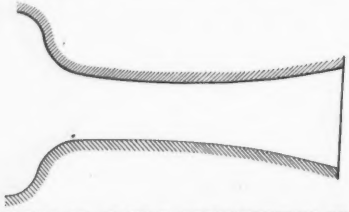


FIG. 3. OUTLINE OF NOZZLE PROPORTIONED FOR VELOCITY DUE TO HEAT DROP

To determine the areas of the nozzle at various points, it is necessary to find the volume of the steam at those points and the velocity which the steam will have, due to its drop in heat.

From the law already stated, the pressure at the throat will be 0.58 times the absolute pressure before expansion, or $p_t = 0.58 \times 165 = 95.7$. It is now necessary, in order to find the velocity V_t at the throat, to use equation (3) which involves the heat before expansion and after. Also we must know the volume per pound of steam at throat conditions in order to determine from equation (5) the area needed at the throat.

The easiest way to find these values is from a Mollier heat diagram, which is shown in Fig. 2, based on Marks and Davis steam tables. For steam at 165 lb. on the saturation curve, the heat as read from the vertical scale is 1195 B.t.u. Dropping to 95.7 along a vertical line, the corresponding heat value is 1152 and the dryness is 0.962. From steam tables, the volume of dry steam at 95.7 lb. is 4.61 cu. ft. per lb., hence $v'_t = 4.61 \times 0.962 = 4.43$ cu. ft.

From (3) $V_t = 224\sqrt{1195 - 1152} = 1465$ ft. per sec. And from (5) the area of the throat will be,

$A_t = 0.225 \times 4.43 \times 144 \div 1465 = 0.098$ sq. in.

For a circular opening, this will give a diameter of 0.3525 in.

For the outlet of the nozzle the pressure, p_2 is 15.7 lb. Dropping vertically from the 165 lb. point on the saturation curve of the heat chart to the 15.7 lb. point, the heat content on the vertical scale is 1025 B.t.u. and the quality is 0.868 dry. From steam tables $v_2 = 25.29$ cu. ft., hence $v'_2 = 25.29 \times 0.868 = 21.95$ cu. ft.

From the scale at the right of Fig. 2 the velocity can be found directly when the heat drop is known. For instance, with the drop for the throat of 1195 — 1152 = 43 B.t.u., the corresponding velocity from the scale is read directly as 1465 ft. per sec. For the outlet, the drop will be 1195 — 1025, or 170 B.t.u. Allowing for 15 per cent friction, the amount producing velocity will be 170×0.85, or 144.5 B.t.u. and the velocity from the scale is 2685 or we can check this by calculation.

From (4), allowing for 0.15 loss,

$$V'_2 = 224\sqrt{(1195 - 1025)(1 - 0.15)}$$
$$= 2685 \text{ ft. per sec.}$$

And from (5) we have

$A_2 = 0.225 \times 21.95 \times 144 \div 2685 = 0.2645$ sq. in.

corresponding to a diameter of 0.579 in.

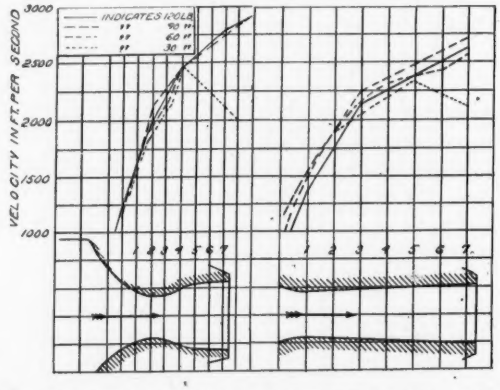


FIG. 4. EFFECT OF SPECIAL NOZZLE DESIGN ON VELOCITY IN NOZZLE

Other diameters between throat and outlet could be determined by this method for pressures between p_t and p_2 and a curve would be found as indicated in Fig. 3 for the section of the nozzle, but experience and experiment have shown such refinement has little practical value, and such a nozzle is difficult and expensive to make.

Kneass made experiments on injector nozzles of various shapes and found that for a straight nozzle as seen in Fig. 4 the velocity during discharge at various points was as shown by the corresponding curves, while for the special curved nozzle, designed so as to give constant acceleration of the jet, the velocity was as shown by the curves above it. While there is some gain in velocity, it is not sufficient to warrant the added expense.

It is, therefore, customary to make nozzles with straight sides having a taper of 1 to 12 which has been found to give best results, and corresponds closely to another rule sometimes given to have the sides at an angle of 5 deg. to the axis. The throat is sometimes made straight for a distance equal to half its diameter.

For the inlet, Rosenhain says that a rounded inlet increases the weight discharged and a diverging outlet affects the velocity of discharge. If the angle of the outlet is correct, it will greatly reduce the vibration

of the jet, hence increase the available energy. But if too large a radius be used for rounding the inlet, more steam will tend to enter than can be passed economically by the throat, and friction will be increased. The usual rounding is to a radius equal to the diameter of the throat, shaping this part of the nozzle by reversed quadrants of circles. Using these practices, Fig. 1 gives the shape of the nozzle for which diameters were calculated.

The length of the nozzle should be such as to expand it to the final pressure desired, but too long a nozzle results in waste by friction, and too short a nozzle causes vibration of the jet after leaving. One formula for the length is:

$$L = 5.715\ (d_2 - d_t),$$

which gives the sides an angle about as previously stated.

To determine the pounds per second which a properly designed nozzle will discharge, Kneass gives the formula:

$W = A_t \times$ lb. per cu. ft. at initial pressure $\times$ 6.19, the value 6.19 being an experimental constant. Or from equation (5):

$$W = \frac{A_t \times V'_t}{144\ v'_t}$$

when p_t is 0.58 p_1

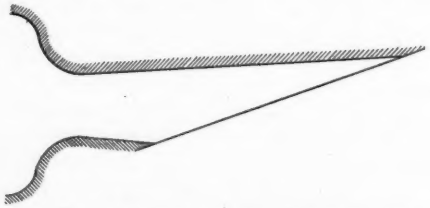


FIG. 5. NOZZLE WITH BEVELED END AND SIDES AT 5 DEG. TO AXIS

Steam per horsepower hour will theoretically be $2545 \div (H_1 - H_2)$, and for a nozzle of good design, the efficiency of the nozzle should be 60 to 80 per cent.

As explained in the section upon arrangement, it is necessary to set the nozzles of axial flow turbines at an angle to the plane of the wheel in order to get entrance and escape of the steam. Practice has settled on an angle of about 20 deg. as the best, the usual angles being 20 or 22.5 deg., and the end of the nozzle is beveled to that angle, to minimize the clearance between nozzle and wheel, giving the form shown in Fig. 5.

In the case of multi-pressure-stage turbines, the value of p_2 is generally more than 0.58 p, so that only the converging part of the nozzle is used, as shown in the views of pressure stage turbines in the commercial descriptions, as Curtis, Terry, Kerr and De Laval.

In reaction turbines, the nozzles are usually of rectangular section, and the pressure drop is such that only converging sides are used, the shape being secured by casting the outlines and hand filing to finish or by bending strips of steel and casting them into the guide rings or diaphragms.

Nozzles in early designs were of cast iron; but this was found unsuitable, as the high heat caused warping and cracking. Bronze or nickel steel are now the common materials, carefully worked to shape and size and highly polished. It should be said, however, that experience has shown that cracked and deformed edges of nozzles in pressure stage turbines have little effect on capacity or efficiency of the turbine, so long as the area through the nozzles is not decreased. This is explained by the fact that heat due to friction in nozzles goes back into the steam, superheating it so that it works more efficiently in later stages.

Nevertheless, it is desirable to have nozzles carefully designed, made and finished, as improper action or excess friction in nozzles will have a considerable effect on the durability of both nozzles and wheels, and will derange the distribution of work among the various stages.

Ash and Fuel Efficiency

IN discussing the general need of fuel conservation in the boiler rooms, Van. H. Manning, Director of the Bureau of Mines, says:

One phase of the present coal situation may put an entirely different light on the supposed increased production of coal of the present year. In round numbers, there was produced 600,000,000 tons of fuel last year. Statement has been made that 50,000,000 more tons will be mined this year, but the preparation of this increased quantity of coal has not been as good as in times past. Analyses of samples show in many cases a greatly increased quantity of ash. Repeated cases are brought to the attention of the Bureau of Mines where coal which would run from 6 to 8 per cent ash in normal times is running from 12 to 18 per cent this year. Complaint about the preparation of coal is general and it is not at all improbable that 5 per cent more ash is included in this year's coal than in previous years.

If this is true, it means that 32,500,000 tons of the estimated output of 650,000,000 tons is nothing but increased ash. If we can imagine over 600,000 car loads of ash being added to the present burden of transportation, the evident effect on car supply and transportation troubles is seen. And this is not all; there is another factor well known to engineers which is apt to be overlooked. Extensive experiments carried on by the Government at the St. Louis Exposition showed that with the coals used there was a decrease of about 1.5 per cent in efficiency for each 1 per cent addition to the ash content of the coal—that is to say, the inclusion of more ash with the coal not only decreases the value of the fuel by the amount of the useless ash, but it makes the remaining good coal less effective to the extent of 1.5 per cent for each 1 per cent of ash. The inclusion of 5 per cent more ash in the fuel, therefore, means a reduction in efficiency of the remaining good coal of about 7.5 per cent, which, added to the 5 per cent useless ash, makes a total reduction in effectiveness of 12.5 per cent.

Although 650,000,000 tons may be produced in 1917, its effectiveness is probably about 7/8 of that of previous years, and equivalent to a production of normally prepared coal of about 570,000,000 tons. We have, then, instead of an increased production as compared with last year, an actual decrease of effective coal of about 30,000,000 tons.

Turbine Blading

Shape and Dimensions for Blading of Impulse and Reaction Types

As already explained, the blades of both types receive energy by impulse and reaction, but for the impulse type, blades are designed to absorb energy by reducing velocity of steam without change of pressure, while reaction blading allows expansion of steam and reduction of pressure in the blading, thus increasing the reaction effect. This involves a difference in shape and angles to secure the best results.

As impulse blading is the simpler, this will be taken up first.

Impulse Blading

In order to get smooth entrance of the steam jet to the wheel, the back of the blade at the entering side is made parallel to the direction of motion of the jet relative to the wheel. This relative motion is compounded of the actual motion of the jet as it issues from the nozzle, and the motion of the wheel blades and is found by means of the parallelogram of velocities.

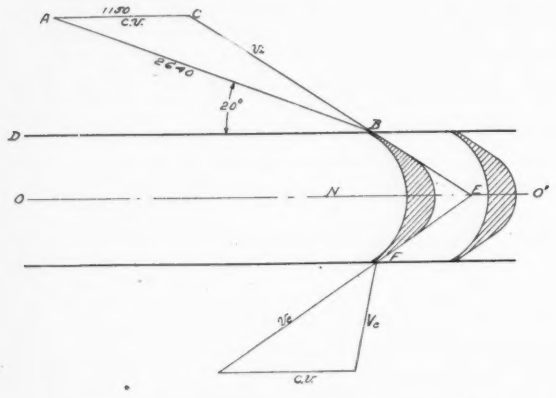


FIG. 1. DIAGRAM FOR DETERMINING SHAPE OF SINGLE STAGE IMPULSE BLADING

Assuming that the jet is delivered at an angle of 20 deg. to the plane of the wheel, and has a velocity V_2' of 2640 ft. a sec., and that the velocity c.v. of the blades is 1150 ft. a sec., we should get the diagram, Fig. 1, where A B represents V_2', A C represents c. v. and B C is the relative velocity, v_1, which by measurement to scale is 1682.5 ft. per sec. and acts at an angle C B D to the plane of the wheel. The back of the blade would, then, have this angle as shown equal to C B D.

To get high efficiency, it is necessary to design the blades so that the actual velocity on leaving the blade V_e is as small as possible, and the angle A B D should be made small. As stated under Nozzles, this angle is usually from 20 to 25 deg. French states that good results are secured by choosing c.v. so that the angle C B D will be twice A B D, but this is modified by mechanical considerations of strength of the wheel construction. Table I gives figures from the practice of the De Laval turbine.

TABLE I. DATA FROM DE LAVAL IMPULSE TURBINE WHEELS

Horsepower	Wheel Diam. In.	R.p.m.	Bucket Velocity Ft. per sec.	Bucket Height In.	Bucket Width In.
5	4	30,000	515	0.6	0.4
10			565		
15	6	24,000	617		
30	8⅞	20,000	774		
50	11¾	16,400	846		
100	19¾	13,000	1115	1.10	0.4
300	30	10,600	1378		
500				1.4	0.5

In practice, c.v. is usually so chosen that angle C B D is from 28 to 36 deg., and for manufacturing reasons the angle of the back of the blade on the exit side is often made equal to that on the entrance side. This also avoids end thrust. In other designs, the sides are made different in order to secure best efficiency. French shows that to get V_e small, the angle of the exit side with the plane of the wheel should be made small,

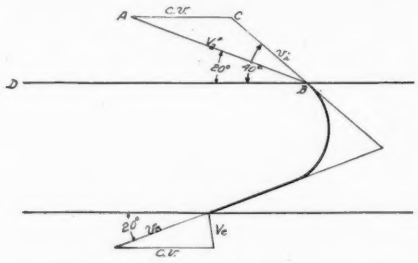


FIG. 2. EFFECT OF EXHAUST ANGLE OF BLADING ON FINAL VELOCITY OF STEAM

and the relative velocity of steam to blade at exit V_e should equal the blade velocity c.v. This would give a blade shape as in Fig. 2.

In some 2-stage Curtis machines of 1500 kw. capacity with 150 lb. gage pressure and vacuum of 1 lb. absolute, running at 900 r.p.m., each pressure stage has four velocity stages, and the entrance angle for first row blades is 20 deg., the exit angle 30 deg.; in the last row blades, both entrance and exit angles of blades are 60 deg. Width of blades in high pressure stage where the pressure is practically atmospheric is 1.3125 in., and pitch 0.75 in. For the first high-pressure row the height of the entering side is 1 in., for the second row 1 5/16 in., for the third row 1 7/16 in., and for the fourth row 1 13/16 in. In the low pressure stage the buckets are 1.625 in. wide, the pitch 1.0625 in. and the heights are for first row 1 5/16 in., for second 1 ⅞ in., for third 2 7/16 in., and for fourth 3 in. Heights at the exit side are made enough greater so that the ends of the buckets form a straight line from one end of a pressure stage to the other. Stationary blades are the same width as mov-

ing blades, and clearance is about 1/16 in. between stationary and moving blades.

Assuming that exit and entrance angles are to be equal, the blade of Fig. 1 is completed as follows: When the width of blade is known, the center line of the blade is drawn dotted, O O′, and angles B E O and O E F made equal. From B and F draw lines perpendicular to B E and F E and their intersection with O O′ establishes the center N for the arc of the front of the blade. The radius of this arc is made 0.97 N B to give thickness to the edge of the blade. Thickness at the center is made about 0.2 the width, if that will give sufficient stiffness to stand the mechanical strains, and for long blades, the thickness at the root is made greater than at the end to give greater strength against bending and centrifugal force. The radius of the back of the blade is made such as to bring the arc through the required point to give proper thickness and come tangent to B E and F E, or it may be drawn concentric to the front of the next blade.

Roe gives as blade widths from 1 in. for 100 hp. up to 1.5 in. for 1000 hp.; the pitch of blades from 0.5 in. to 1 in. and the height of the first row 1.55 the diameter of the nozzle at exit.

Actually it is desirable to take account of friction on the bucket in determining the exit angle, and it is found that this will reduce the velocity about 10 per cent. If we draw the velocity diagram at exit on Fig. 1, we find that the resultant of $v_e = 0.9\ v_1$, and c.v. = 1150 ft. per sec. is V_e which will be the actual velocity of the steam as it leaves the blade and shows the direction in which it would pass away from the wheel.

Efficiency of action of the blading is found from the expression, Eff. $= (V'^2_2 - V_e^2) \div V'^2_2$.

Multi-Velocity Stage Blades

For multi-velocity staging, part of the velocity is extracted by each set of moving blades, permitting of a lower blade velocity. For two wheels, the blades will run at half the velocity of a single wheel, or ¼ the velocity of the steam jet; for three wheels at ⅓ the velocity of a single wheel or 1/6 that of the jet and so on. To illustrate the method of determining the shapes of successive blades, assume an initial pressure of 150 lb. absolute and 145 deg. superheat, which will give a total temperature of 500 deg. F. Condenser pressure to be 1.5 lb. abs. and nozzle angle 22.5 deg. The expansion is to be in three pressure stages and two velocity stages to be used per pressure stage. Allow for the usual 10 per cent loss in each set of blades and 10 per cent in each set of nozzles.

On the heat chart shown in Fig. 2 in the article on nozzles, locate the initial condition; the heat content is 1270 B.t.u. Part of the chart is reproduced in Fig. 3 for convenience. Dropping vertically to the 1.5 lb. line, the heat content is 948 or a drop of 322 B.t.u. For each pressure stage, it would be ⅓ of this or 107 B.t.u. Heat content at the end of the first stage would be 1270 — 107 = 1163, and at the end of the second stage, 1163 — 107 = 1056, and corresponding pressures are 45 and 10 lb.

For the first nozzles, $107 \times 0.9 = 96.3$ B.t.u. converted into velocity energy giving 2175 ft. per sec. velocity, and the bucket velocity would be $2175 \div 4 =$ 544 ft. per sec. For mechanical reasons, this may well be reduced to 400 ft. per sec., which will be used as c.v. and the first blade form is found as for a single stage wheel, the inlet velocity, v_1, being 1815 ft. per sec., as seen in Fig. 4. For the exit velocity, v_e, we would have 1815×0.9, or 1633 ft. per sec., which, combined with c.v. of 400, gives 1290 ft. per sec. as V_e and its angle determines the angle for the stationary reversing blades. Design of these blades is completed as for the moving blade. There is 10 per cent loss in the guide blade, hence the exit velocity is $1290 \times 0.9 = 1161$. Combining this with c.v. = 400 we have 865 as v_1 for the second velocity stage blades, and its angle determines the form of the second stage blades, which are completed as for previous blades.

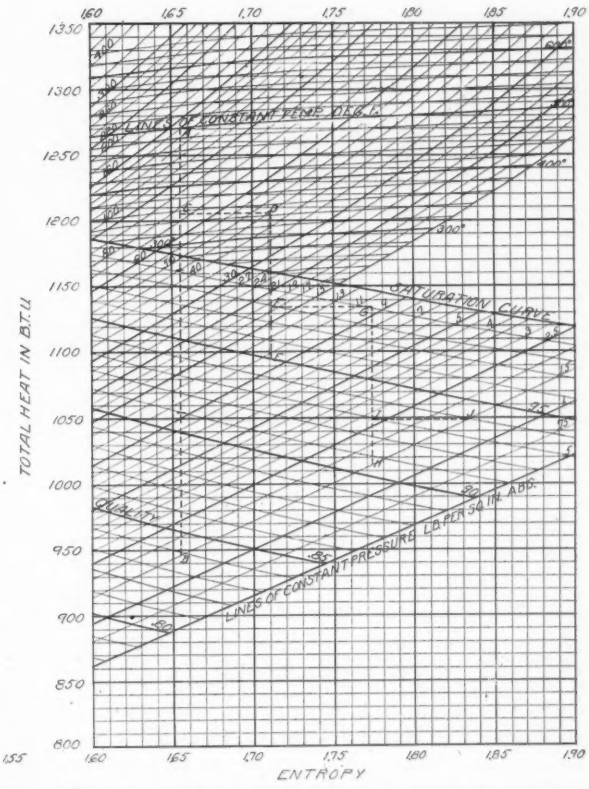


FIG. 3. HEAT CHART USED IN DESIGN OF MULTI-STAGE BLADING

For v_e from the second stage blades, we have $865 \times 0.9 = 780$ ft. per sec., which combined with the 400-ft. c.v. gives 615 ft. per sec. as V_e for the second stage.

To figure the work done per second by a wheel, we multiply the sum of the component of the jet velocity in the direction of blade motion, v_1', and the component of the exhaust velocity in the same direction v_{11}' by the blade velocity and divide by 32.16.

For the first wheel we have:

$$\text{Work} = (v_1' + v_{11}') \times \text{c.v.} \div 32.16$$
$$= (2010 + 1050) \times 400 \div 32.16$$
$$= 38{,}060 \text{ ft. lb.}$$

Likewise for the second wheel:

$$\text{Work} = (v_{111}' + v_{1111}') \times 400 \div 32.16$$
$$= (940 + 90) \times 400 \div 32.16$$
$$= 12{,}811 \text{ ft. lb.}$$

or a total of 50,871, or, say, 51,000 ft. lb. of work in this pressure stage. Dividing by 778 gives 65.6 B.t.u. as the equivalent of this work. The heat content will then be 1270 — 65.6, or 1204.4 B.t.u. Dropping from the initial condition at 150 lb. and 500 deg. to 1204 B.t.u., Fig. 3, and tracing horizontally to the 45-lb. line, gives the condition of the steam at the beginning of expansion for the second pressure stage, showing that it is still superheated and at a temperature of 345 deg. F. Dropping from that point to the 10-lb. position, the heat content is 1095 B.t.u., a loss of 119 B.t.u. $119 \times 0.9 = 107.1$ B.t.u. converted into velocity energy in the second set of nozzles, which for the heat-velocity scale would result in 2320 ft. per sec. velocity.

By the same method as for the first stage, and using 400 ft. per sec. as c.v. of the blading, the blade forms would be worked out, and we should get velocities as follows:

	First Wheel	Second Wheel
Entrance, actual.....	2320	1220
Entrance, relative.....	1890	925
Exit, relative.....	1698	835
Exit, actual.....	1360	650

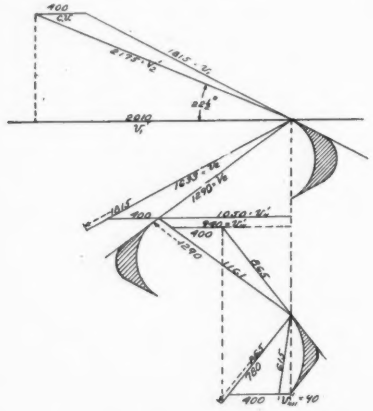


FIG. 4. DESIGN OF TWO VELOCITY STAGE SYMMETRICAL IMPULSE BLADES

Taking components in the direction of motion of the blades and calculating the work as before, this becomes about 53,500 ft. lb. a sec., or the equivalent of 68.8 B.t.u. This deducted from 1204, the heat content at the beginning of the second stage gives 1135 as heat content at the beginning of the third expansion. Following this heat course from D to F, Fig. 3, and over to the 10-lb. position at G gives the condition of the steam, which is found to be 0.99 dry.

Dropping from G to the 1.5-lb. line at H shows a heat remaining of 1012 B.t.u., or a drop of 1135 — 1012 = 143 B.t.u., and the velocity corresponding is 2670 ft. per sec.

Working out conditions for the blading in the third pressure stage by the same method as for the first two, we find the work to be equivalent to 86.5 B.t.u., which leaves 1049 B.t.u. in each pound of steam at final exhaust. Locating this value on G H at I, Fig. 3, and tracing to the 1.5-lb. line gives J as the final condition of the steam, 0.94 dry, instead of 0.84 dry, which would have been the condition at B if there had been no friction effect.

Calculation shows that the combined blade efficiency of a turbine of capacity 300 to 3000 kw. is from 60 to 70 per cent; for smaller sizes, 50 per cent down to 10 kw. It will be noted that in Fig. 4 blades of succeeding stages are of different shape, and the last one is quite flat, which tends to cut down its efficiency. Some designers prefer to make the guide blades unsymmetrical in order to get a sharper angle of entrance for moving blades of the second and following wheel blades. Such a construction is shown in Fig. 5, initial conditions being the same as for Fig. 4, but the guide vane being designed to give an entrance angle of 22.5 deg. to the jets for the second wheel.

Reaction Blading

As already explained, blading in the reaction type has two functions. At the entrance to absorb energy by impulse action, and on the exit side to take up energy by reaction and from expansion of the steam. The shape is, therefore, entirely different from that for impulse wheels. As a rule, no nozzles are used. A set of guide blades directs the steam against the blades in the first stage, and succeeding guides, reverse the

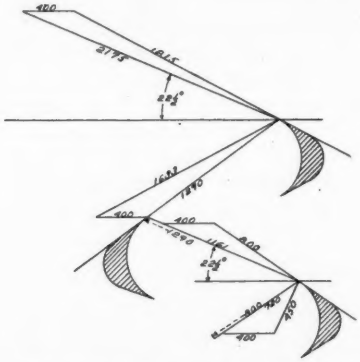


FIG. 5. TWO-VELOCITY STAGE BLADING FOR EQUAL ENTRANCE JET ANGLES

direction of the steam, allow of further expansion to increase velocity and direct the steam at the desired angle against the succeeding wheel blades. The actual velocity of the steam increases continuously from stage to stage, and both wheel blades and guide blades increase in length to furnish area for the passage of the increasing volume of steam. Blades are mounted on drums of several diameters. When the steam velocity has increased to from 2 to 3.5 times that of the moving blades, the diameter of the drum of the rotor is increased and a new series of blade lengths is started. Guide and moving blades have the same general form, the exit angle being made 20 to 30 deg., which thus becomes the actual angle of the entering jet of steam.

It is customary to make the blade velocity c.v. from 0.3 to 0.6, the jet velocity and the drums of the rotor are proportioned to have a constant ratio of diameters from step to step. Pressure drop in each stage is small, so that jet velocities are lower than in impulse turbines, which results in flatter entrance sides to the vanes, and lower blade velocities, 115 to 130 ft. per sec. in the first stages. Short blades are used in the high-pressure

stages to avoid end leakage by giving small clearance, but for mechanical constructive reasons, the blades are seldom less than ¼ in. high. For the larger drums in low-pressure stages, the blade velocities are made as great as the strain on blade attachment will permit.

The use of a small exit angle gives greater pressure drop per stage, but produces long, narrow channels which increase friction and cause eddy currents. The angles already mentioned, 20 to 30 deg. are chosen as a compromise.

As to blade speeds common in practice, Roe gives the following table from a paper by E. M. Speakman.

TABLE II. BLADE PRACTICE IN PARSONS' TURBINES

Rated Capacity kw.	Blade Velocity Ft. per sec. First Expansion	Last Expansion	No. of Rows	R.p.m.
5000	135	330	70	750
3500	138	280	75	1200
2500	125	300	84	1360
1500	125	360	72	1500
1000	125	250	80	1800
750	125	260	77	2000
500	120	285	60	3000
250	100	210	72	3000
75	100	200	48	4000

To show the process of blade form determination, an example taken from Harding and Willard on Mechanical Equipment of Buildings will serve.

Dry steam is to be taken at 165 lb. absolute, exhaust as 1 lb. absolute, a heat drop of 326 B.t.u., the rotor to have three steps or barrels. Assume blade velocity as 135 ft. per sec. for the first barrel, 220 for the second and 330 for the third, and exit angle of blades as 22.5 deg.

If the ratio of blade velocity to steam velocity be taken as 0.6, the latter, for the first barrel, will be $135 \div 0.6 = 225$ ft. per sec., and the velocity diagram for the blade will be as in Fig. 6. The components of initial and final steam velocities parallel to the blade motion are 209 and 73 ft. per sec., hence the energy abstracted by one row of blades will be $135\ (209 + 73) \div 32.16 = 11.88$ ft. lb., and the heat drop per stage, $11.88 \div 778 = 1.52$ B.t.u. If 25 per cent be allowed for added heat drop due to friction and leakage, the actual will be $1.52 \div 0.75 = 2$ B.t.u. practically, and if all stages were on the small cylinder the number needed would be $326 \div 2 = 163$ stages. For equal work by the three barrels, the first one would have $163 \div 3 = 54$.

As guide blades and moving blades have the same form, and the same amount of expansion is allowed in each, the shape for one blade determines that for all which have the same blade velocity. The straight part of the exit side is made long enough to clear the point of the adjacent blade, as indicated in Fig. 6. The entrance side has the angle of the relative velocity of the steam as for impulse blading, and the two lines are joined by an arc of a circle. The front of the blade is worked in as a smooth curve leaving sufficient metal for strength requirements.

In practice, the work done by the first barrel will be considerably less per stage than that by the second or third barrels, but the working out of exact relations is a somewhat complicated process, and can be found in detail in either Roe or Moyer on The Steam Turbine.

Construction of Blading

In some makes of reactive turbines, the blades are generally cut, punched or pressed into proper form from strips of drawn material. Some builders use Sankey's solid foundation rings held in place by a soft metal calking strip and, as this form is usually all machine-made, it is considered by many safer than where dependence is placed on hand work, such as must ordinarily be done where each distance piece is calked separately.

Impulse blades are stamped from sheets, drop forged or milled from solid bars with or without a wide base to act as a distance piece or are made from extruded metal strips of desired cross section. Usually these blades are of crescent sections, but some are formed of flat material and made of constant thickness over the width of the blade. The separate distance pieces are usually of the same material as the blades themselves.

In general, all impulse blades are provided with shrouds to prevent vibration and to provide an enclosed

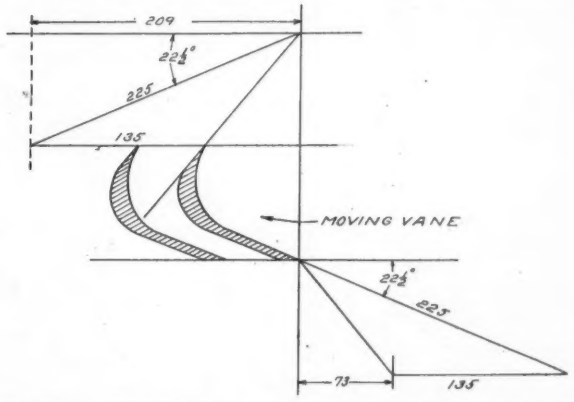


FIG. 6. DETERMINATION OF FORM OF REACTION BLADING

passageway for the steam at high velocities. As there is no drop in pressure between the two sides of a row of moving blades, the clearance may be made large, both on the end and sides, so that there is little possibility of rubbing when in operation.

Impulse blading is usually held in place in dovetailed grooves or inverted T-shaped slots, although some manufacturers form their blades with two legs which straddle the disks and are held firmly in place by rivets.

Builders who employ blading on which the discharge side of the blades makes a sharper angle with the axis than the inlet side, to reduce the velocity of exhaust have the blade usually lengthened radially on the discharge side, so that inlet and discharge areas are made equal.

The City council of a foreign government wishes to receive bids for an electric light plant according to plans and specifications, for public and private lighting and for the sale of electric light and energy. Bids will be received until March 20, 1918, which is the farthest limit. Copies of the proposal and all information may be had by writing to the Bureau of Foreign and Domestic Commerce, mentioning No. 25,941.

Turbine Wheels

IN THE wheels, the stresses are due to centrifugal force, to vibration from unbalancing, and to unbalanced steam pressure. The third is not usually of great moment, for in impulse wheels, the pressure difference between the two sides of the wheel is negligible, and, as blades are nearly symmetrical, the end thrust on them is small. In reaction turbines, the blades are carried on drums rather than disks so that only the blades are affected by the difference in pressure between stages.

Vibration effect is reduced as much as possible by careful adjustment of the wheels before and after assembling, to get them in both static and running balance. Some vibration will be present, however, and the wheels must be stiff enough to withstand this.

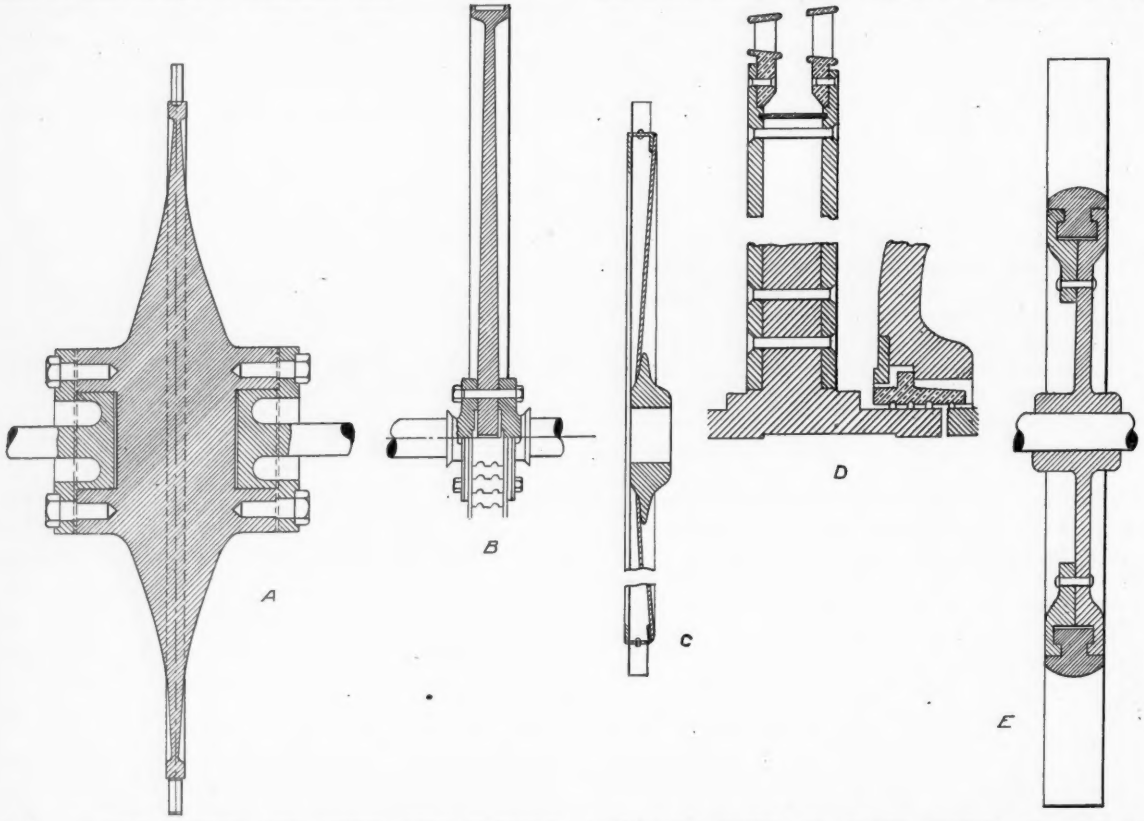


FIG. 1. A—SOLID TAPERED DISK WHEEL WITH FLEXIBLE SHAFT
B—SOLID DISK WHEEL WITH BOLTED-ON HUB
C—WHEEL OF PLATES BOLTED TO HUB
D—HOLLOW WHEEL OF PLATES RIVETED TO HUB AND DISTANCE PIECE AT RIM
E—SOLID DISK AND HUB WITH RIM SPLIT AND RIVETED UP TO CLAMP BLADING

Centrifugal force depends on the weight and speed of the moving mass, and as the speed is great at the outer edge of the wheel, the force per pound of mass is great. Each part of the wheel must, however, withstand the force of all the wheel outside it, so that the total stress is greater nearer the shaft than at the rim, and the body of the wheel must increase in strength from rim to hub. At the rim, extra metal must be provided for attachment of the buckets, which increases the centrifugal force, but is unavoidable.

Wheels are either solid, or built up of a hub and plates, several types being shown in Fig. 1. Others are illustrated in the descriptions of commercial machines.

Drums of reaction turbines are usually cast hollow, either as one piece or in sections, according to size. In larger sizes, the drums proper are hollow cylinders carried by disks or spiders which are mounted on the shaft, while in smaller sizes, the spider and drum are made as a single piece. In either case, the drum form gives a maximum of rigidity, and the design is made to secure strength, balance and a support to the blading.

Forms of blading are best seen in the illustrations of commercial machines, and methods of attachment to wheels and drums are in many cases given in detail. The most usual fastening is by a dovetail blade root fitting into a slot and made fast, either by calking in a wedge strip, by bolting up a separate clamping strip on the rim or by a driven fit in the slot. Blades are steadied by a lacing through the center, a shroud ring fitting over the ends, which are riveted, or both. In some designs the outer end of the blades carries a projecting lip, which abuts against the one adjacent, thus forming a solid ring. This, or the shroud ring, serves to prevent escape of steam past the ends of blades, and to prevent catching of the buckets and consequent injury in case of shifting of the wheel which allows contact with the casing or guide rings.

Riveting of blades to rims is indicated in Fig. 1 C, cross dovetailing in the illustration of De Laval blading, blading sections riveted to rims in Fig. 1 D, dovetailed slots in drums in Allis-Chalmers blading, and buckets milled from solid rims in the Sturtevant and Terry wheels. The Kerr wheel has blading cast into the rim.

In wheels that are to run at very high speeds, it is necessary to take special precautions for strength. A hole in the center reduces the strength of a solid disk by half, hence for large sizes one builder uses a solid disk with recessed hubs cast on the sides and bored out with a slight taper. The shaft ends are enlarged and turned taper to fit the recesses, then bolted in place. As an added precaution against serious accident a groove is turned just inside the rim so that if breakage of the disk occurs it will be in this groove, and sections of the comparatively light rim will let go, saving the heavier disk from breaking. Such pieces will probably be stopped by the casing, and the unbalanced force will cause rubbing of the hub on the casing sufficient to act as a brake and slow down the wheel below the speed to do further damage.

Blade Rings

RINGS attached to the casing for controlling the steam are used for carrying nozzles or guide blades. In the first stages the nozzles are set singly or in groups covering part of the circumference, and in the lower stages the nozzles frequently cover the entire rim of the wheel. Guide blades are arranged similarly for the impulse type, but in reaction turbines the guide blading extends around the entire circumference for all stages.

Examples of single nozzles are the De Laval and the Terry arrangement. Curtis uses the grouped nozzle, and Parsons the entire reaction blade ring.

Methods of holding the blades vary according to the ideas of designers. Some are held in dovetailed grooves in the casings, having the inner ends with or without shroud rings. Some are fastened into inner and outer rings, the latter setting in a groove in the casing. Some are made up into segments of rings which are fastened to the casing or to diaphragms which extend in close to the shaft. The details of these can best be studied in the descriptions and illustrations of commercial machines, but some typical arrangements are here shown.

Figure 1 shows the arrangement of nozzles used by Rateau and Curtis, each set somewhat wider and set somewhat in advance of the preceding set to take care of the increased volume of the steam and its forward motion in the wheel blades. In later stages the nozzles would cover the entire rim of the diaphragm.

In Fig. 2 is shown the single nozzle arrangement used by De Laval, which is similar to the method followed by Kerr, Terry and Sturtevant, each nozzle being independently hand controlled.

Forms of blade diaphragms in section are shown in Fig. 3, A being the Rateau design, B that of Zoelly, C that for Kerr machines. For reaction types, the forms of blade rings are as shown, with the descriptions of the Westinghouse and Allis-Chalmers makes.

In impulse types there is small tendency to leakage past the guides, as there is no drop in pressure through them and the tendency of the steam is to follow a straight course through the blading. For pressure staging, it is customary to use a solid joint, packed or ground, between stages at the outer circumference of the nozzle diaphragms, and a labyrinth or carbon ring packing at the shaft. The drop is small between successive stages, so that leakage is not great.

In reaction turbines there is pressure drop at each row of blades, either stationary or moving, but this drop is small for each row. Tendency to leakage is greater in the first stages since the end clearance needed mechanically is a larger percentage of the blade length. This clearance must be larger for high blade velocities, hence it is usual to allow 115 to 130 ft. per sec. blade

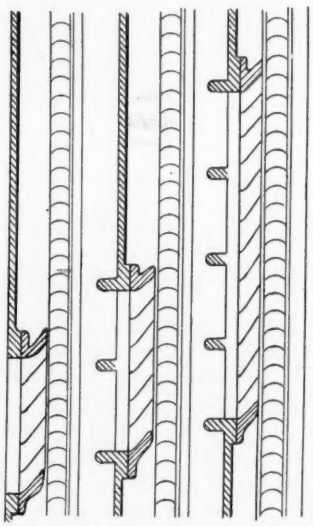


FIG. 1. ARRANGEMENT OF GUIDE BLADE SECTIONS

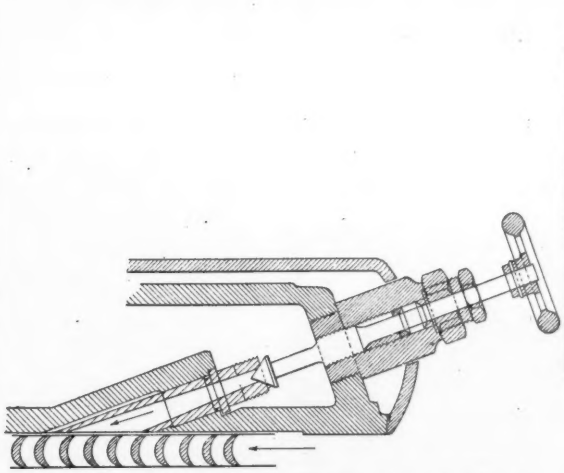


FIG. 2. SINGLE NOZZLE WITH HAND CONTROL

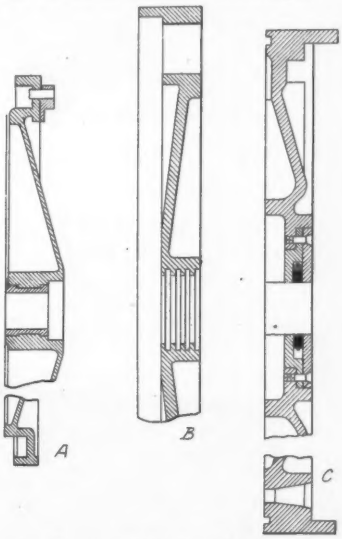


FIG. 3. A. RATEAU DIAPHRAGM. B. ZOELLY DIAPHRAGM. C. KERR DIAPHRAGM

velocity in the first stages and keep the blades as short as possible, but seldom under ¼ in. high. The clearance is then made close, between blades and casing for the moving blades, and between blades and drums for the stationary blades. Often a shroud ring over the end of the blades permits of even closer clearance and if of channel form, it tends to give a labyrinth effect and further check leakage.

Single nozzles are usually bored from a casting which is bolted to the casing or to a diaphragm. Groups of nozzles are cast as a block and worked to shape by hand, or are formed by casting curved steel blades into the block or directly into the diaphragm rim. For complete rings of nozzles or blades, the blading may be cast into sections of the diaphragms or set into grooves and a shroud ring attached by projections from the blade ends which are riveted through the shroud rings.

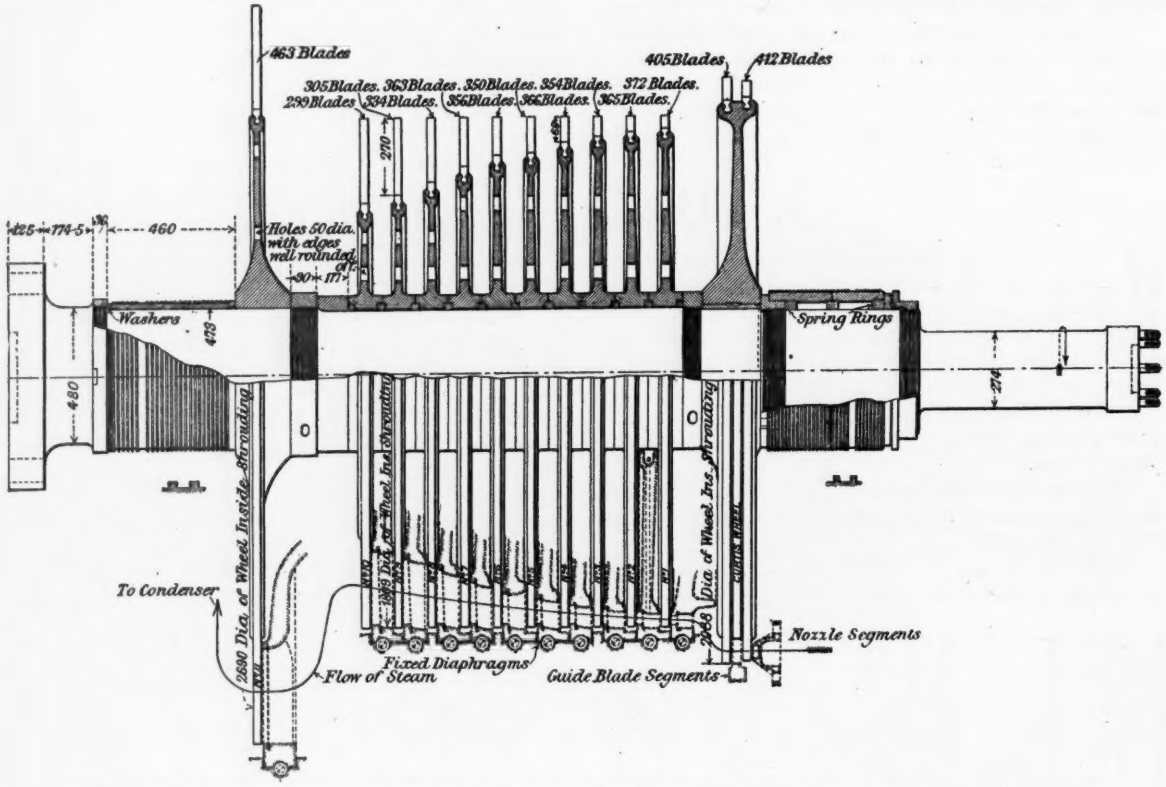


FIG. 4. SECTION OF PART OF TURBINE SHOWING ARRANGEMENT OF HIGH AND LOW-PRESSURE SECTIONS

The important points are to secure the blades against change of position, to reduce leakage, and to guard, so far as possible, against stripping of blades in case distortion results in contact of stationary and moving rings.

Axial clearance between stationary and revolving blades is frequently made half the blade width in the reaction type, and for impulse wheels runs from 0.06 in. for a 500-kw. unit to 0.1 in. for a 5000-kw. unit. Clearance at blade tips is small, one rule being that the clearance shall be 0.01 in. per foot of rotor diameter + 0.01 in., but in any case not less than 0.02 in. Allow-

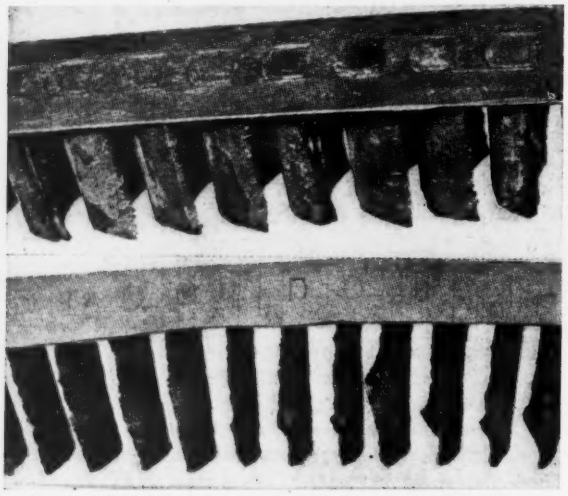


FIG. 5. A. ALUMINUM BRONZE WHEEL BLADING AND SHROUD AFTER 24,000-HR. RUN. FIRST HIGH-PRESSURE WHEEL B. BLADING FROM SECOND WHEEL

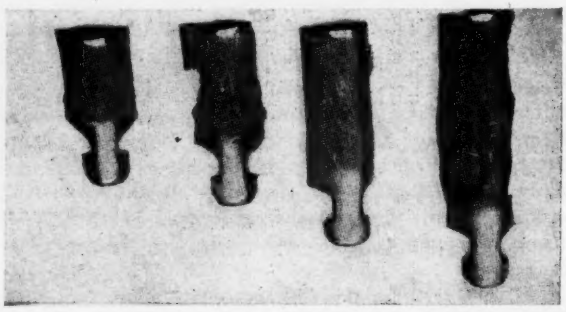


FIG. 6. BLADES FROM LOW-PRESSURE STAGES AFTER 33,660-HR. RUN

ance must be sufficient to take care of such deflection as is liable to take place when the casing is heated to running temperature.

Some Experience With Blading

Interesting facts were given in regard to nozzle and blading performance by A. Fenwick before the South African Inst. of Engineers. It should be noted that the machines from which these figures were obtained were all of German manufacture, and as pointed out by Frederick Samuelson in Engineering, the failures were probably due to use of improper materials rather than type of machine or design.

These turbines were of impulse type with a high-pressure two-velocity stage and 11 lower pressure single-velocity stages, Fig. 4. The nozzles were cast-iron blocks, and blading of 5 per cent nickel steel, of aluminum bronze, of brass, nickel brass, monel metal, 25 per cent nickel steel, 30 to 32 per cent nickel steel and Siemens-Martin steel.

The record shows that for the aluminum bronze, the aluminum soon disappears from the surface, leaving a brittle layer of copper which cracks and flakes off, the process being repeated until the blade splits up. Often the part riveted into the shrouding chips off, letting the shroud ring loose. Figure 5 A shows the condition in

FIG. 7. SECTION OF CAST-IRON NOZZLE BLOCK

the first row of the high-pressure stage after 24,000 hr. running, and 5 B is from the second row.

These blades have shorter life than those in the low-pressure stages, probably due to the higher temperatures encountered, but blades below the fourth single velocity stage show erosion and pitting of the backs, and edges become thin and easily broken, as in Fig. 6. This roughness seemed to have little effect, however, on steam consumption, as tests before and after reblading showed no material improvement due to the new blades.

Cracking of the longer aluminum bronze blades at the root where they were attached to the wheel by a tee head was also common, although the stress from centrifugal force was not excessive.

Steel blades, while standing up well, show some erosion at the inlet edges on the low pressure stages

From these results, it appears that nickel steel stands up best for blading under high temperatures, aluminum bronze is satisfactory for lower stages if vibration is not too great, and brass gives good results, but is probably not as durable as the bronze. For the low stages, erosion due to the wet steam makes the advisability of nickel steel doubtful.

Inlet nozzles on the high-pressure end were of cast iron, with section as shown in Fig. 7; the form is circular at the inlet and rectangular at the outlet, where the edges are thin. These thin edges seem to grow under the action of superheated steam, cracking badly; and, frequently large pieces of V form break out and pass through the turbine. Little damage has been done except a slight bruising of the blades, but there is always liability of serious trouble. Test showed that the nozzles with pieces cracked out gave as good steam economy as when in good condition, hence in putting in new nozzle blocks, a V-shaped section was cut away, and a saw cut run in at the end of the notch to allow for expansion, as shown in Fig. 8. Although there is still some cracking, pieces do not break off. In another nozzle block, steel blades, forged to the shape of the cast-iron vanes, were cast into an iron body. After 3000 hr. running, these were found as good as new.

For the stationary blading, aluminum bronze proved unsuitable; 25 per cent nickel steel stood up well for a time, but became brittle with use. Five per cent nickel steel stands up well, although edges of the first stationary row become thin and brittle after a time. The life is some 20,000 hr. of running. Brass, used for the second and third rows, stands up well, the maximum run being 30,670 hr.

For guide blading in the low-pressure stages, 30 to 32 per cent nickel steel was used, and shows varying and inconsistent results. The first sign of trouble is fine surface cracks, forming a network beginning on the

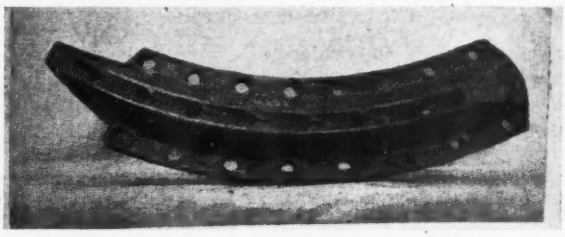


FIG. 8. NEW NOZZLE BLOCK WITH NOTCHES CUT TO RELIEVE EXPANSION

inlet side of the blade. Finally pieces come loose and pass through succeeding stages severely damaging the wheel blading. The trouble sometimes starts in isolated blades, sometimes in groups of blades, but all eventually crack, and unless given attention the guide ring collapses. Sometimes the vanes break away from the outer retaining ring, sometimes the break is in the middle of the blade. Even when the blades seem to be in good condition, a light tap with a hammer will cause the cracks to appear.

Several months elapse between the forming of the first cracks and breaking away of pieces, so that examination at intervals of 3000 running hours will show when guide blading should be renewed in turbines having guide rings. In diaphragms having a solid center, there was a tendency to movement of the diaphragms towards the following wheel as soon as the cracks began to appear, probably because the blade pitch was greater in this construction than where guide rings were used.

Little deterioration was found in the first four stages, although there was some pitting and corrosion. The blades here were of ordinary steel containing no nickel. In stages 7 and 12, which had blades of 30 per cent nickel steel, cracks appeared under the hammer test, but there was little corrosion. At the high-temperature end of the turbine, the cast iron of rings and diaphragms "grew" or took permanent expansion set under the action of the superheated steam, so that the bosses of

the dished diaphragms moved towards the preceding wheels. But this movement was not sufficient to take care of all the expansion, and the blades in the first two rings buckled considerably, since the outer edges of rings and diaphragms was held in a groove in the casing. In addition to buckling, the blades are cracked as shown

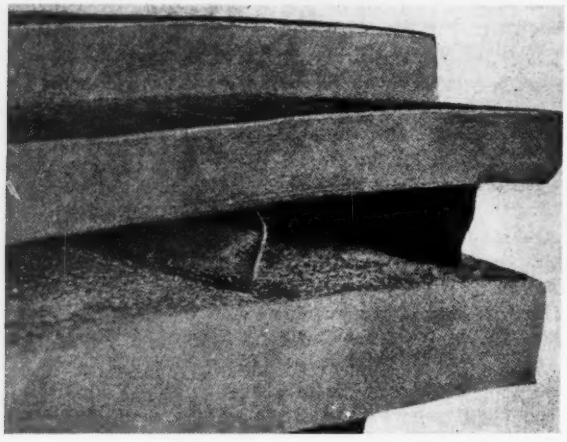


FIG. 9. EXAMPLE OF DISTORTION OF GUIDE VANES DUE TO EXPANSION OF DIAPHRAGM

in Fig. 9, which shows part of a ring after 39,000 hr. use. This effect is only in the higher-pressure stages, buckling seldom being found beyond the third guide ring.

The stress was relieved by machining a 0.12 in. clearance between the guide ring and diaphragm, resting the diaphragm on two short-fitting strips on the lower half, and fitting a few copper pins on the upper half of the guide ring with about 0.02 in. clearance between pins and diaphragms, so that the latter cannot rise enough to foul the shaft. These pins are eased periodically, as examination shows that they are being crushed by the diaphragm.

For the guide blade solid diaphragms, which are suspended by side lugs, in some, 0.12 in. clearance was left all around, but expansion sidewise gradually caused binding in the grooves. When this was eased, there was no further trouble. In other machines, where only 0.04 in. clearance was left in the grooves, this was taken up and in some cases heavy distortion of blades occurred after a year's run; the inner part of the diaphragm moved in some cases as much as 3/16 in. toward the following wheel, and practically every blade was cracked.

On the other hand, some rings have shown a long life without being eased, but the steam temperature was somewhat lower, about 550 deg. F. (290 C.) as against 600 to 680 deg. F. (320 to 350 C.) in those machines that gave trouble. If cast steel were used for diaphragms, this growth and distortion would probably not occur.

Consideration of results seems to indicate that medium carbon steel is a better material for guide blading than 30 per cent nickel steel, although experience with the former has not been of sufficient length to be conclusive.

Shafts, Bearings and Lubrication

PREVENTING VIBRATION AND LEAKAGE

IN any machinery which runs at as high a speed as turbines, unbalancing, which is difficult to eliminate entirely, results in tendency to vibration, and either the shaft must be made flexible, as is done in smaller sizes of the De Laval, so that it can spring and allow the rotor to rotate about its real center of gravity, or the shaft and bearings must be made heavy enough to resist the vibrating tendency and hold the rotor reasonably steady. In the first method, the bearings are made movable so that the shaft can spring. In the second, the shaft is made stiff and as short as possible, bearings have spherical seats, and in some cases concentric sleeves are used with small oil holes through them so that the shaft may move sidewise slightly, changing the thickness of the oil film between sleeves, and thus adjusting itself to the natural position of rotation, as in Westinghouse bearings.

To get a stiff shaft requires large diameter, which increases the tendency to steam leakage along the shaft where it passes through the casing and between pressure stages, so that no larger shaft should be used than is needed for rigidity.

Balancing to reduce vibration is accomplished by keeping weight distribution as uniform as possible, by testing assembled wheels, and the entire assembled rotor, and adding weights or drilling out metal to counteract any lack of uniformity. Static balance is usually tested by placing the assembled rotor on horizontal knife edges, rolling it along, and noting the part that settles to the bottom, and the force required to change that position.

Running balance is tested by running the rotor at moderate speed and noting the high spots, which indicate the parts of the rotor that are heavy. Evidently any weight added or metal removed to effect balance should be opposite the high spot both on diameter and along the axis. If this is not done, a couple will be introduced when running which will cause vibration that may become serious.

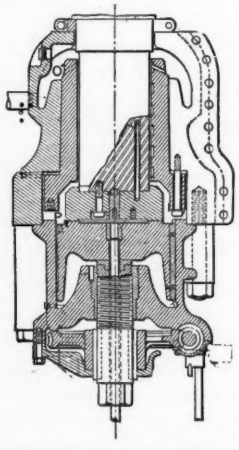
For impulse wheels, the axial length is small compared to the diameter and good static balance will usually produce reasonably good running balance, and for reaction turbines, the drum speed is lower so that slight dynamic or running unbalance does not have great effect. For large units, the generator field will have more effect in producing vibration than the turbine rotor, and the great weight makes the balancing on knife edges difficult. A method that has been sometimes used is to drop the rotor and field into the bearings that have been freshly oiled, and note what part settles to the bottom. This must be done quickly, as friction will interfere with the turning of the rotor if it is allowed to stand so that the oil film is squeezed out of the bearing.

Wheels and drums are held to the shaft by keys, forced fits or compression couplings or by a combination of these. In the case of drums, spiders are often mounted on the shaft and the drum is fastened to the spiders.

Bearings are usually split horizontally, or vertically, ring oiled with ball seats to allow the bearing to line itself to the shaft, although in some machines a straight bearing is used and wedges or adjusting screws provided to permit of careful alining. Thrust bearings are made up of disks mounted on the shaft and a grooved bearing, usually babbit lined, which is adjusted endwise to hold the rotor in proper position for the correct clearance between moving and stationary blades. For the larger sizes, the bearings are provided with water jackets to reduce heating. Also oil is forced through, usually by a pump driven from the turbine shaft to ensure ample lubrication.

In vertical turbines the step bearing which carries the weight of the rotor consists of a cast-iron block, splined and doweled to the the lower end of the shaft, resting on a second block of cast iron or lignum vitæ, a recess is turned in the face of the lower block to which oil or water is supplied under a pressure of 180 to 450 lb. by a pipe connection through the center of the block. The lubricant drains out between the blocks, thus carrying the weight on a film of oil or water at all times. The lubricant is filtered and passes to a reservoir to be recirculated.

To reduce leakage of steam along the shaft two forms of bearing are in use; the labyrinth has a succession of rings running in grooves to give a long path, and usually on the high-pressure end has provision for carrying off the steam from some groove to a low-pressure stage of the turbine; on the low-pressure end, steam is supplied to the labyrinth at moderate pressure, to avoid in leaks of air, which would be troublesome in the condenser. A second packing is the carbon ring, which is a close fit on the shaft, having several rings in series, placed in separate chambers. These are frequently used in combination with the labyrinth type, or with a centrifugal wheel packing such as shown in connection with Allis-Chalmers turbines.



PARTS OF STEP BEARING FOR VERTICAL TURBINES

Packing

Grooves and rings in labyrinth packings are not meant to touch, as this would cause friction; the radial passages are made narrower for the inflow than for the outflow in order to counteract the velocity by centrifugal action as much as possible. From 12 to 24 grooves are used, one-third of these being a steam or water seal. The grooves are sometimes made in the shaft and brass rings attached to the casing by screws. In other designs, disks are turned on the shaft and grooves left in the casing.

Balancing pistons are packed against leakage by similar rings and grooves. Clearance allowed is from 0.007 to 0.02 in. The number of grooves used may be made 1.3 times the diameter of the piston in inches for the high-pressure; 0.66 times the diameter for intermediate, and 0.25 times the diameter for low-pressure piston.

Rings are usually of brass ½ by ⅛ in. fitted into grooves in the casing and either held by a dovetail, or made a loose fit so that they can revolve. Grooves are turned in the periphery of the pistons.

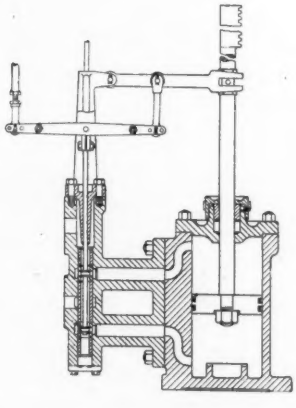
Governing of Turbines

Classification of Methods

GOVERNING is usually accomplished by one of four methods or a combination:

1. Throttling of steam as it enters the steam chest.

2. Varying the area for admitting steam to the turbine, usually by varying the number of nozzles in action.



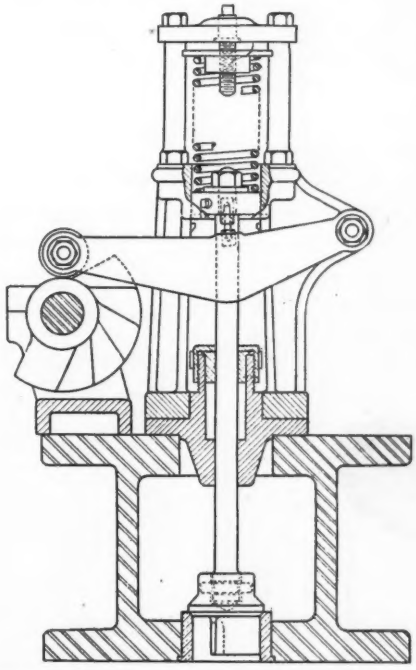


FIG. 1. DETAILS OF CURTIS "HYDRAULIC" CONTROL OF NOZZLE VALVES

3. Varying the time during which steam is admitted to the steam chest, used mostly on Parsons type and known as the "blast" method of regulation.

4. By-passing steam to lower stages, employed frequently for carrying overloads.

Overspeed governors are really safety stops, and act by tripping a valve to close off admission, or in case of condensing turbines by breaking the vacuum at some point.

Throttling has the disadvantage of reducing the possible efficiency of the turbine, since steam flows to the turbine chest at a less pressure than that in the mains. This is somewhat offset by a slight superheating of the

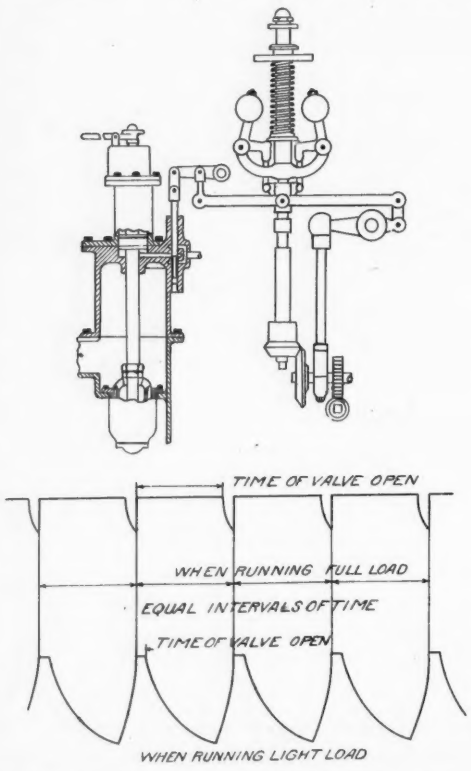


FIG. 2. MECHANISM OF WESTINGHOUSE GOVERNOR CONTROL AND DIAGRAM SHOWING EFFECT ON STEAM ADMISSION

steam, reducing the friction in nozzles and blades; but this gain is not sufficient to make up for the loss. If, however, the load is fairly constant so that the throttling is not great, the throttling method gives good results in

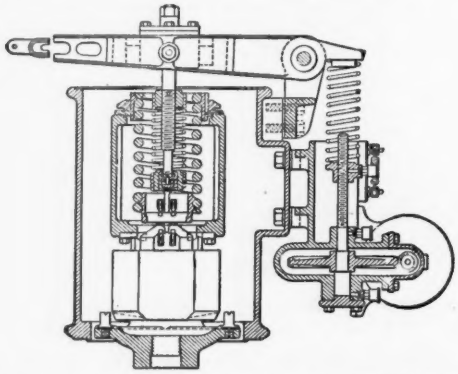


FIG. 4. SECTION OF CURTIS GOVERNOR SHOWING AUXILIARY SPEED CONTROL SPRING

practice, and its great convenience makes it a favorite, especially for machines of small capacity.

In several makes, the throttling control by the governor is combined with hand control of the number of nozzles in action, thus keeping the throttling effect small, even for light loads; but this is not feasible where there are wide and rapid fluctuations of load, except to a limited extent. This method is found in the Kerr Economy Turbine, the De Laval, the Terry and the Sturtevant.

Control by the governor of the number of nozzles in action is commonly used on impulse type machines, the Curtis being an example.

Three methods have been used. In the first, the governor actuated a drum on which electric contact strips were mounted in helical arrangement. These controlled circuits leading to solenoid magnets which in turn actuated pilot valves to operate steam pistons on the valve stems. The valves were held closed by springs and opened by the pistons, the number of valves held open corresponding to the demand on the turbine.

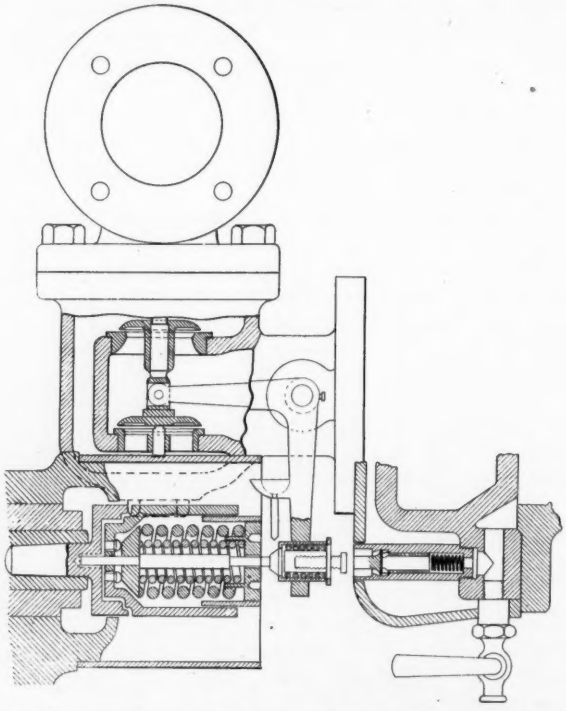


FIG. 3. DETAIL OF DE LAVAL GOVERNOR WITH THROTTLE VALVE AND OVERSPEED VACUUM BREAKER VALVE

In the second method, now mostly used for turbines up to 1500 kw. the valves are operated mechanically. An eccentric driven from the shaft by worm and gear gives motion to a rocker arm through a bell crank, and on the rock shaft are mounted supports and pawls, one for opening and one for closing each valve. Cams, on a second shaft whose piston is regulated by the governor, hold the pawls out of engagement with ratchets on the valve stems or allow engagement so as to open or close the valves, so that the nozzles in operation will furnish the steam needed to carry the load.

The third method, used for larger units, has the valves, operated by a piston and rock shaft with oil pressure, the piston being locked in position when stationary by oil at each end, and motion to open or close valves being regulated by a pilot valve controlled by the governor. The valves are opened by cams on a rock shaft and closed by springs. The cams are set in helical series so that valves open in succession, Fig. 1.

Formerly a number of valves were used to give fine gradation as to the nozzle areas in service; but experience has shown that better results in regulation are secured by the use of fewer nozzles, and the present tendency is to use not over four, the intermediate supply being regulated by partial opening of the valves producing some throttling of the steam.

Overload is cared for in some cases by a set of nozzles admitting live steam to the second pressure stage, these opening only after all first-stage nozzles are wide open and the speed has fallen appreciably. But this involves complication in the governor gear, and it is better to design the turbine to carry its greatest allowable load on the first stage nozzles and run it somewhat underloaded for the lesser outputs.

Varying the time of admission, or intermittent blast is well illustrated by the governor of the Westinghouse turbine. The main valve is opened and closed at a rate of 150 to 250 admissions a minute by an eccentric and a

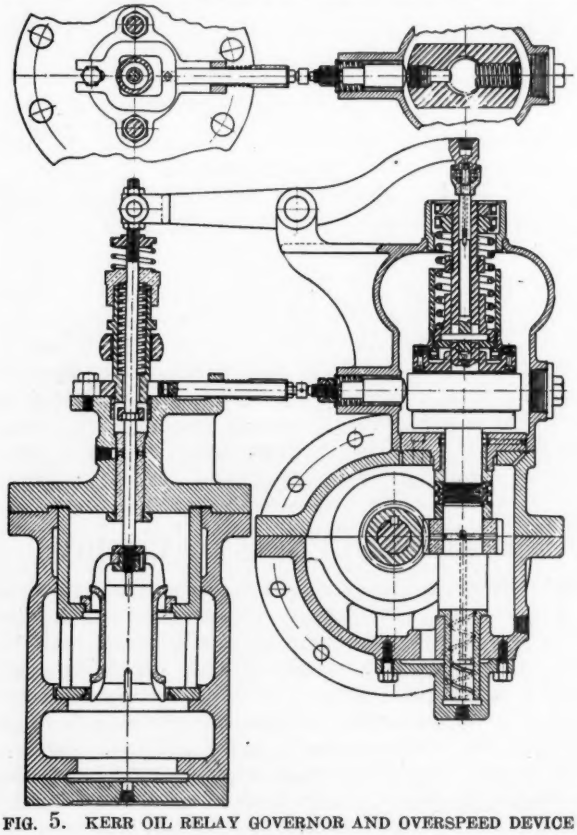


FIG. 5. KERR OIL RELAY GOVERNOR AND OVERSPEED DEVICE

rocker arm, Fig. 2, moving a pilot valve. This valve in turn controls the pressure on a piston on the valve rod, and the duration of admission is regulated by the position of the rock shaft of the compound pilot valve lever. The governor sets this rock shaft so as to give an admission varying from an instant for light loads to practically continuous flow for overloads. The valve is in constant motion and can never stick; but at very light loads there is chance for cooling down of the turbine between puffs to an extent that may prove undesirable, Fig. 2.

Bypassing is in two forms. One has a series of passages leading to the first, second and third barrels of the rotor, and these are uncovered in succession by the main valve as the load increases. A second has an overload valve, which is only opened when speed has fallen unduly, and then admits steam to the second barrel. It is more frequently used with the composite type, which has a high pressure impulse wheel and reaction blading for the low pressure. It is illustrated in Westinghouse and Allis-Chalmers descriptions.

Practically all governors are of a modified flyball type, acting against springs whose tension can be regulated to control the speed, Fig. 3. For turbines driving a.c. generators, this spring tension is usually arranged to be adjusted by a small motor controlled from the

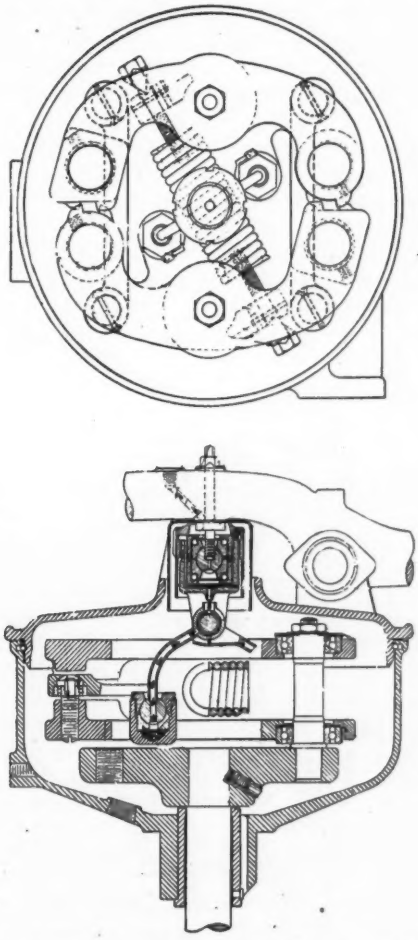


FIG. 6. VIEWS OF CURTIS INERTIA GOVERNOR

switchboard to aid in synchronizing when bringing the unit into service. The Curtis governor has the speed regulated by means of an auxiliary spring on a rod connected to the lever which actuates the nozzle control mechanism, its tension being regulated by a handwheel or by a motor, as desired, Fig. 4.

Overspeed governors are frequently mounted on the end of the shaft, or, as in the Kerr turbine, on the spindle of the main governor. When the speed exceeds the desired limit, the overspeed trip is thrown, releasing a spring which closes an auxiliary steam valve, as shown for the Allis-Chalmers turbine, or the main steam valve, as for the Kerr turbine. In the larger De Laval machines,

the overspeed device opens an air inlet into the casing, by the means shown at the right of Fig. 3, permitting part of the wheels to revolve in an air cushion.

In the smaller horizontal Curtis turbines the nozzle admission is controlled by a series of valves in a cylindrical chamber. A spindle runs through the center of this chamber and carries centers which are so machined that they open the valves in succession as the spindle is moved lengthwise by the operating cylinder under control of the pilot valve. These nozzle valves are of double seated poppet type, free to move. They are closed by gravity and steam pressure, and opened by the centers on the spindle.

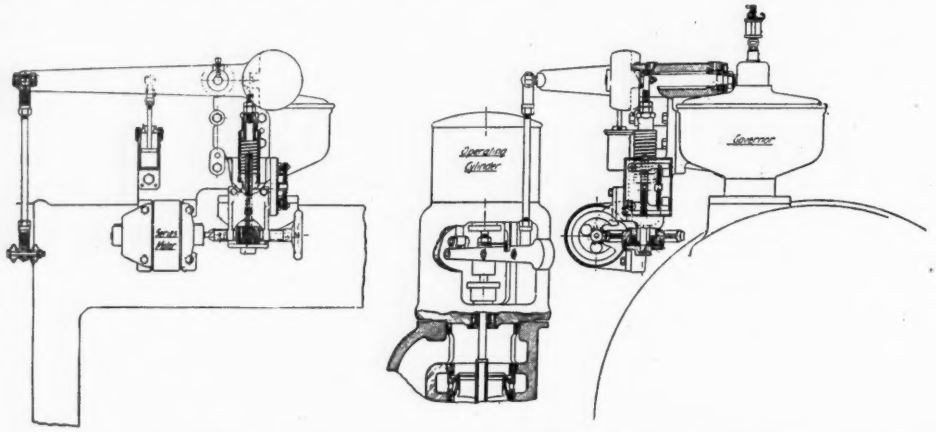


FIG. 7. ARRANGEMENT AND CONNECTIONS OF GOVERNOR FOR SMALL CURTIS TURBINES

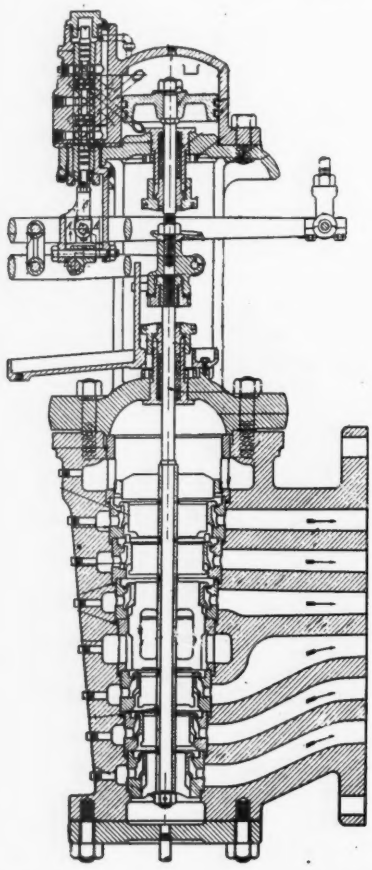


FIG. 8. SECTION OF CONTROL CHAMBER, VALVES, OPERATING CYLINDER AND PILOT VALVE FOR SMALL CURTIS TURBINES

When starting, the piston of the operating cylinder is raised by steam bled to the under side of the piston, thus opening all nozzle valves. When the governor arms of combined inertia and centrifugal type move outward, its end of the governor lever is moved upward, and moves the pilot valve downward, admitting steam above the operating piston, shifting this downward and closing nozzle valves until the steam admitted is that needed to maintain the speed set by the governor. As the piston and valve spindle move downward, the pilot valve is moved upward until it comes to neutral piston, when motion ceases until the governor again moves the pilot valve.

Turbine Cylinders or Casings

THESE are usually of cast iron and are shaped to give the least clearance outside the wheels that is consistent with mechanical safety. They must be so proportioned as to stand the stress due to internal pressure, but this is not great except at the high-pressure end. In the horizontal arrangement, the plain casting is used, the cylinder being in upper and lower halves; but in the vertical arrangement, especially for large sizes, the cylinder is usually made in sectors and ribbed, flange joints being provided for bolting together in assembling. At the high-pressure end, a steam chest chamber is provided in the form of a partial or complete ring, usually as a part of the cylinder casting, and in this steam chest are mounted the nozzles or nozzle castings to which steam is admitted—by valves in those machines which have variable admission area governing. Exhaust is into a chamber at the low-pressure end, which is formed by a ring enlargement of the cylinder casting. Where the steam is admitted to a part only of the high-pressure wheel, the admission should be arranged at several points symmetrical about the axis in order to get uniform heating of the casing and guides so as to avoid distortion from uneven expansion.

Many smaller capacity, single-pressure machines are so arranged that the entire main cylinder or casing forms the exhaust chamber, the steam chest being cast as a ring on the outside of the casing, thus reducing the stress on the casing and the tendency to leakage along the shaft.

Cylinders are cast separate from the bed plate and bolted down by feet, but even where the turbine and its driven machine are bolted to a single bed plate and connected by a flexible coupling, it is necessary to test the alinement when installing, as speeds are so high that even a small error in adjustment, or displacement during transit, is liable to result in severe stress of the shafts.

In smaller sizes the shaft bearings are carried on brackets bolted to the casing and are not likely to get out of line. In larger sizes separate pillow blocks are employed and must be lined up as carefully as engine bearings.

Casings for small impulse type machines are often made of solid ring form, the exhaust end, and in some makes the steam-chest end, being made as a plate which bolts to the side of the ring. This is a strong design, but loses something of convenience in case of repairs, since one bearing must be removed in order to get at the turbine wheel.

Inside the cylinders are turned smooth and have dovetailed grooves for the insertion of guide blades direct, or are grooved to receive the guide-blade rings. One make has the cylinder made in ring sections, each carrying a set of nozzles and having a flange that forms the wheel chamber. The flanges are fitted together with tongue and groove packed joints, giving a steam-tight enclosure.

Most joints are, however, a ground fit, bolted iron to iron, and calked, if necessary, to secure tightness. The fewer and simpler the parts, so long as the shapes do not become difficult to cast and machine, the less will be the expense to manufacture and the likelihood of trouble in operation.

Sufficient metal to insure easy and successful casting will usually give the strength needed to resist steam pressure, and the common cylindrical form is the one best suited to resist such pressure; but in case of flat surfaces, it is well to be certain that sufficient ribbing is provided to prevent deflection of any consequence. A maximum stress of 2000 lb. per sq. in. is as great as should be allowed in the metal.

Deflection of the cylinder is due more to heat effect than to weight. In some designs the passage connecting the exhaust chamber to the low-pressure balance pistons, which are at the high-pressure end, is made a part of the casing; this tends to cool the lower part of the casing so that expansion of the upper part will make the casing spring upward at the center. The side flanges for casings split horizontally influence the expansion, giving an oval form. Attachment of steam and exhaust piping at the lower part of the casing tends, by weight and expansion of pipe, to cause distortion of the casing.

These effects must be guarded against. The balance piston connection may be made outside the casing with provision in its length for taking up expansion. Ribs may be cast top and bottom, or, better, distributed about the circumference to resist distortion and give uniform expansion. Steam connection is often made at the top of the casing, and the weight of the governor carried there to counteract tendency to curve upward. Exhaust connection usually has an expansion joint to take up stress. These matters must usually be worked out by experiment, but there will always be some distortion, and careful tests should be made with the turbine at running temperature to make certain that the distortion is not sufficient to cause trouble.

To avoid unlooked-for strains, a safety valve may well be provided on the exhaust chamber to prevent rise in pressure there in case of blades stripping. Also sufficient drains of exhaust chamber and casing should be provided so that water cannot accumulate in starting up, to cause trouble from blades stripping. The casing drains should be connected to the air pump with shutoff valves, so that water may be withdrawn rapidly when warming up, and the drains closed after running conditions are attained.

At the exhaust end, in order to keep back pressure low and avoid pockets, the area of the exhaust passage should increase according to the volume of steam to be passed. As the connection is usually at the bottom, this would mean that the exhaust chamber should increase in cross-section area from the top downward; also, to reduce back pressure, it is well to increase the section of the exhaust connection as it goes toward the condenser so that the steam velocity at the condenser is half that at exhaust from the rotor. This would be an expensive refinement, however, and where the condenser is close to the turbine, as is usually the case, may not be warranted by the gain to be secured.

Passage at the balancing piston should have an area to carry 5 to 10 per cent of the exhaust steam.

To allow freedom of expansion, the casing is fastened to the bed plate, in case of long turbines at only one end, the other end being free to move on planed ways. Sometimes a support is provided the entire length of the casing, but this may produce uneven expansion, and separate support of the ends is more common. Usually the low-pressure end is bolted fast to the bed plate and the high-pressure end allowed to move with expansion, thus avoiding disturbing the connection to the condenser and the coupling to the driven shaft.

NOTICES ARE DISPLAYED in German machine shops, giving hints on economy in lubricants which should be valuable in any country. These are:

Use only closed cans, with spouts that will deliver drops, or at most only a thin stream.

Use all lubricating apparatus strictly according to the instructions, and put the oil only where it will actually lubricate. If a machine has automatic droppers, shut off the supply while machine is standing.

Do not use cylinder oil on shafting or elsewhere when cheaper oil will answer.

Keep all rubbing surfaces in good condition. Rough surfaces and too-tight boxes consume more oil. Worn and leaky bearings waste oil.

Always use drip pans and arrange to filter and cleanse the oil so caught. It is as good as new.

Collect all greasy waste and wiping cloths, so that the oil may be recovered. Never burn them.

Be careful about using lubricating oil for cooling a bearing. Water will often do as well.

Turbine Foundations

TYPES; AND SETTING UP OF TURBINES

THESE may be of all steel, which gives the maximum of space in the basement for condensing machinery; of concrete and steel, which gives somewhat less basement space, but avoids much of the tendency to vibration; or of all concrete, which is the most desirable, especially for large units.

stantial support and absorbs all tendency to vibrate. For vertical turbines the concrete type is always used.

Size of foundations depends on soil, distribution of weight of turbine and, in case of steel construction, the allowable deflection. The manufacturer should always be consulted and his advice followed, as experience is the only guide as to the allowance needed to prevent settling or distortion that would throw the machine out of line.

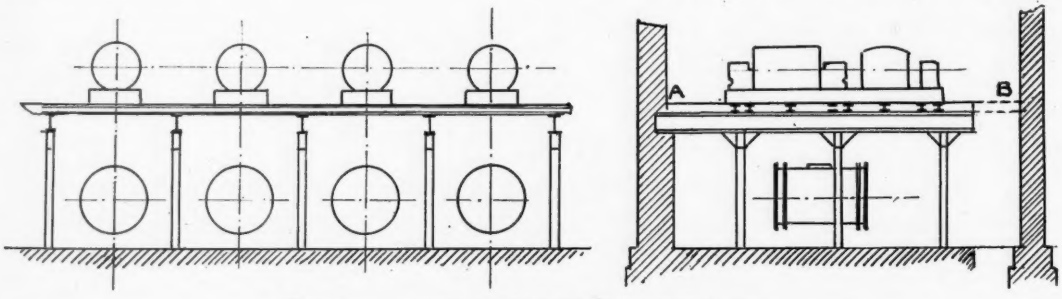


FIG. 1. ALL-STEEL TYPE OF TURBINE FOUNDATION

In the steel construction it is common practice to grout the girders into the building walls as at A, Fig. 1, but it is better to keep them free, as at B. to avoid transmitting vibration to the building. Figure 2 shows the construction for concrete and steel, which to a degree avoids the transmission of vibration from one machine to another or to the building.

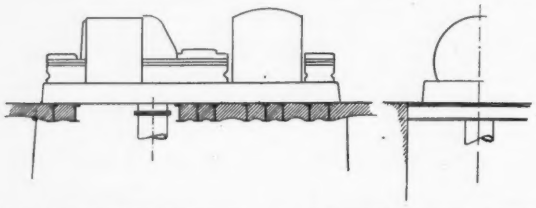


FIG. 2. SECTION OF STEEL AND CONCRETE FOUNDATION

Concrete, as shown in Fig. 3, occupies more space and necessitates placing some of the auxiliary apparatus before the foundation is completed. But it gives sub-

Correct alinement is of utmost importance, and is not difficult to secure. Large machines are shipped in sepa-

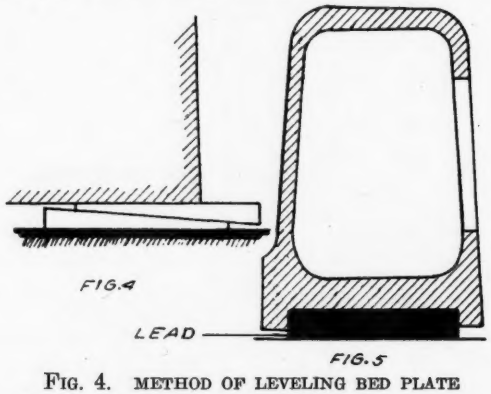


FIG. 4. METHOD OF LEVELING BED PLATE
FIG. 5. USE OF LEAD FOR GROUTING BED PLATE

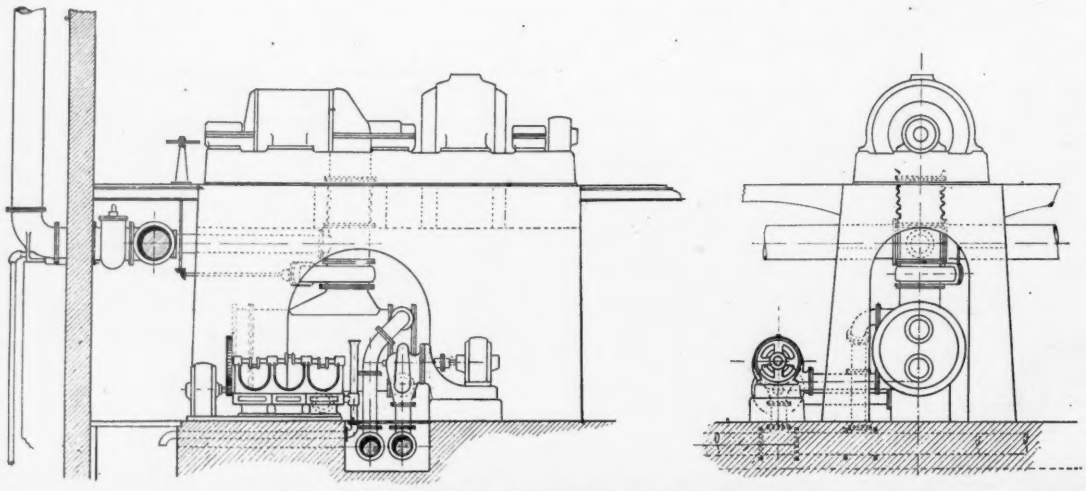


FIG. 3. ARRANGEMENT OF SOLID CONCRETE FOUNDATION

rate parts, the bed plate being placed first on the foundation and lined up level from the machined surfaces in both directions, using plenty of double wedges, as in Fig. 4, to insure that there is no warping of the plate. These wedges should give about ½ in. to 1 in. between plate and foundation and be spaced about 1 ft. apart. Leveling should be to within 0.001 in. in 12 in., and after leveling, foundation bolts should be set up snug and alinement rechecked.

If the foundation is solid concrete, the bed plate may now be grouted in place, but if of steel, it is better to put in place as many parts of the turbine as possible,

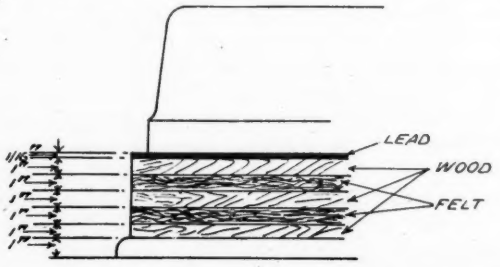


FIG. 6. CUSHIONING BED PLATE TO ABSORB VIBRATION

so that the girders may take their final deflection before grouting. With steel foundations, grouting is often done with lead, as in Fig. 5, to help dampen vibration.

If cement is used, a mixture of 1½ parts sand to 1 of cement is recommended. If spaces are very small, 2 cement to 1 of sand will fill better. Moisten the top of the foundation before grouting; provide a dam so as to back the grout as high as possible, and be sure that the grouting runs up well inside the plate. After grouting, retighten foundation bolt nuts and recheck alinement.

Where it is especially important to avoid noise and vibration, cushions of lead, wood, felt and rubber, or a

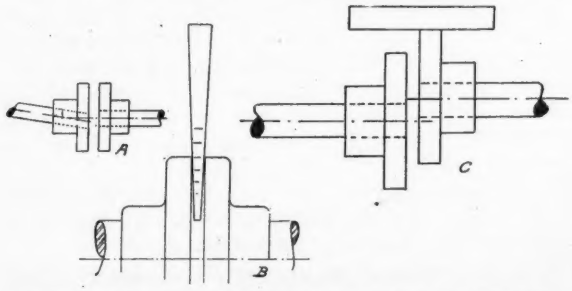


FIG. 7. LINING UP TURBINE AND DRIVEN SHAFTS BY COUPLING

combination of these in layers is used under the bed plates. A 1/16-in. layer of lead is placed next the plate, and 1-in. layers of other materials placed below to a thickness of 4 to 6 in. In such cases no holding-down bolts are used, and special care must be taken that no strain comes on the turbine from piping connections.

Lining of turbine and generator or pump shaft is generally done by the flanges of the couplings, using a straightedge to make sure that the edges are parallel, and micrometer or taper gage to test for parallel faces. Couplings should first be tested to make sure that they are true on the ends of the shafts, to avoid the condition of Fig. 7 A; then test to aline the shafts, Fig. 7 B, and make sure that they are at the same height, Fig. 7 C. As the turbine shaft will rise somewhat with heating of the turbine casing, it is best that the alinement of turbine and driving shafts be done with the turbine hot. Test for evenness of edges and parallelism of faces of couplings should be made with the coupling in four positions by turning the shafts, first one, then the other, through 90 deg. in opposite directions.

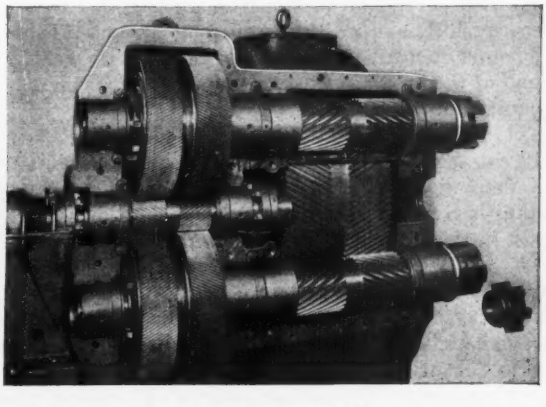
To give an idea of the size of foundations needed, the following tables of dimensions will serve:

DIMENSIONS OF TURBO-GENERATOR UNITS

Capacity kw.	Length Ft.	Length In.	Width Ft.	Width In.	Height Ft.	Height In.	Floor Space Sq. Ft.	Weight Lb.
400	18	11	3	11	5	3	7.4	.....
1000	29	11	5	3	8	4	157	.....
5500	47	3	14	0	14	0	662	.....
Parsons type—60 cycle.								
300	23	4	4	0	5	1	93.2	36,500
500	24	6	4	0	5	1	98	40,000
750	27	0	5	10	5	9	157	65,200
1000	29	0	6	10	6	6	198	81,500
1500	31	10	6	10	6	6	218	103,000
2000	34	9	9	2	8	2	319	138,000
3500	35	9	10	6	9	4	374	237,000
5500	46	0	11	41	0	6	521	417,000
Parsons type — 25 Cycles.								
300	23	6	6	4	5	9	149	51,000
500	24	6	6	4	5	9	155	56,600
750	28	3	6	4	6	0	179	75,500
1000	29	9	7	6	6	9	223	104,200
1500	32	6	7	6	7	0	244	131,000
2000	38	0	9	0	8	0	342	166,000
3500	42	5	11	8	10	5	495	356,000
5500	46	10	13	2	11	6	617	460,000
7500	50	10	13	3	11	6	673	511,000
Parsons type—turbine only.								
75	10	6	2	6	5	0	26.2	.....
150	12	2	3	3	6	0	39.3	.....
250	14	3	3	6	7	0	50	.....
500	17	7	6	9	7	0	119	.....
1000	19	5	8	0	9	6	155	.....
1500	20	7	9	6	9	6	196	.....
2000	20	7	9	6	10	0	196	.....

Reducing Gearing

FOR driving many kinds of machinery, particularly in the case of direct-current generators and with single-stage impulse type turbines, the speed needed to get efficient use of steam are far too high to permit of direct connection, and the ratio of reduction is too great to allow of successful belt driving. Gears have, therefore, been used; but because of the high speed and high ratio, special forms are necessary.



DOUBLE REDUCTION GEARING FOR 1750 HP. TURBINE UNIT

Requirements are that each tooth shall carry but a small load in order to get short, fine pitch teeth; that end thrust be eliminated; that side thrust be small in order to reduce pressure of the fast-running shaft on bearings.

The most prominent design is that of the De Laval Turbine Co., which consists of two pinions, right and left helical teeth of fine pitch mounted on the turbine shaft. These run between two sets of gears in an oil-tight casing, so that each driven shaft takes up half the power of the turbine. The driven shafts may be geared externally to the same generator, or may drive separate machines, as desired.

Pinions are of high carbon steel and gears of mild steel, the pitch velocities being as high as 100 ft. per sec. in the smaller sizes. Up to 30 hp. the gears are cut from the solid, but for larger sizes a cast iron center is used with steel rims, in which the teeth are cut, shrunk on.

By this construction the pressure between teeth may be kept small, as several are in action at once; the right and left spirals neutralize end thrust, and the two sets of gears take up side thrust, so that the high velocity is rendered possible. The gears run in a bath of oil and show but little wear after long periods of service. The spiral is cut at an angle of 45 deg. to the axis, the sizes of gears running from a pinion 1.08 in. diameter, with 21 teeth, and gears 10.1 in. diameter, with 208 teeth of 0.15-in. pitch for 10-hp. units up to pinion of 2.65 in. diameter, with 31 teeth and gears of 29.29 in. diameter, with 362 teeth of 0.26-in. pitch for 300-hp. units.

Other makers use gears of similar type, especially for marine work, where the teeth in one design have an angle of 20 deg. to the spiral and run at 1900 ft. per min., pitch velocity.

Another design of gearing made by the General Electric Co. uses right and left helical teeth, with the pinion driving two sets of gears. The double gear is built up of flexible disks in each half, these disks being keyed to the shaft and bolted together at the center. Near the outer surface an axial clearance of 0.01 in. is left between disks, the webs being cut away to allow axial flexure, so that excessive tooth pressure exerted by the pinion at any point will produce flexure of the disk and relieve this pressure. The laminated construction is shown in the illustration which is of a set for a 1750-hp. turbine, reducing from 3200 to 137 r.p.m. In this set, pinions on the gear shafts drive a double helical gear on the propeller shaft of a ship.

Turbine Economy

Effects of Variation in Steam Pressure, Superheat, Vacuum and Size Upon Economy of Steam Turbines

IN a discussion of turbine economy two phases of the subject must be considered, first, by what means is it possible to secure higher efficiency; second, is the expenditure for the necessary means and their operation warranted, considering the gain in efficiency obtainable.

Reports of turbine tests give efficiencies calculated from the results obtained and it is essential that this be clearly defined in order that their significance may be appreciated.

Thermal efficiency is the heat output, equivalent to the work done, divided by the heat input. Considering a complete turbo generating unit, to produce 1 kw.-hr. we must use a certain number of pounds of steam which is called the water rate (W R). The actual heat units used per pound of steam in making this transfer of energy is $H_1 - q_2$, H_1 being the total amount of heat in one pound of steam at the throttle and q_2 is the heat in a pound of the liquid at exhaust pressure. Expressed as a formula, the thermal efficiency is:

$$E = 3412 \div (H_1 - q_2)WR$$

3412 is the heat equivalent of one kilowatt hour. The determination of the thermal efficiency is valuable in that it gives an idea of the great amount of heat that is lost in this method of conversion into work.

The ideal efficiency is the thermal efficiency represented by adiabatic expansion of steam under the given conditions and the heat convertible by a turbine working on adiabatic expansion is $H_1 - H_2$, H_1 being the total heat in 1 lb. of steam at the throttle and H_2 the total heat in 1 lb. of steam expanded adiabatically to exhaust pressure. The efficiency of a turbine working under this ideal condition is $(H_1 - H_2) \div (H_1 - q_2)$.

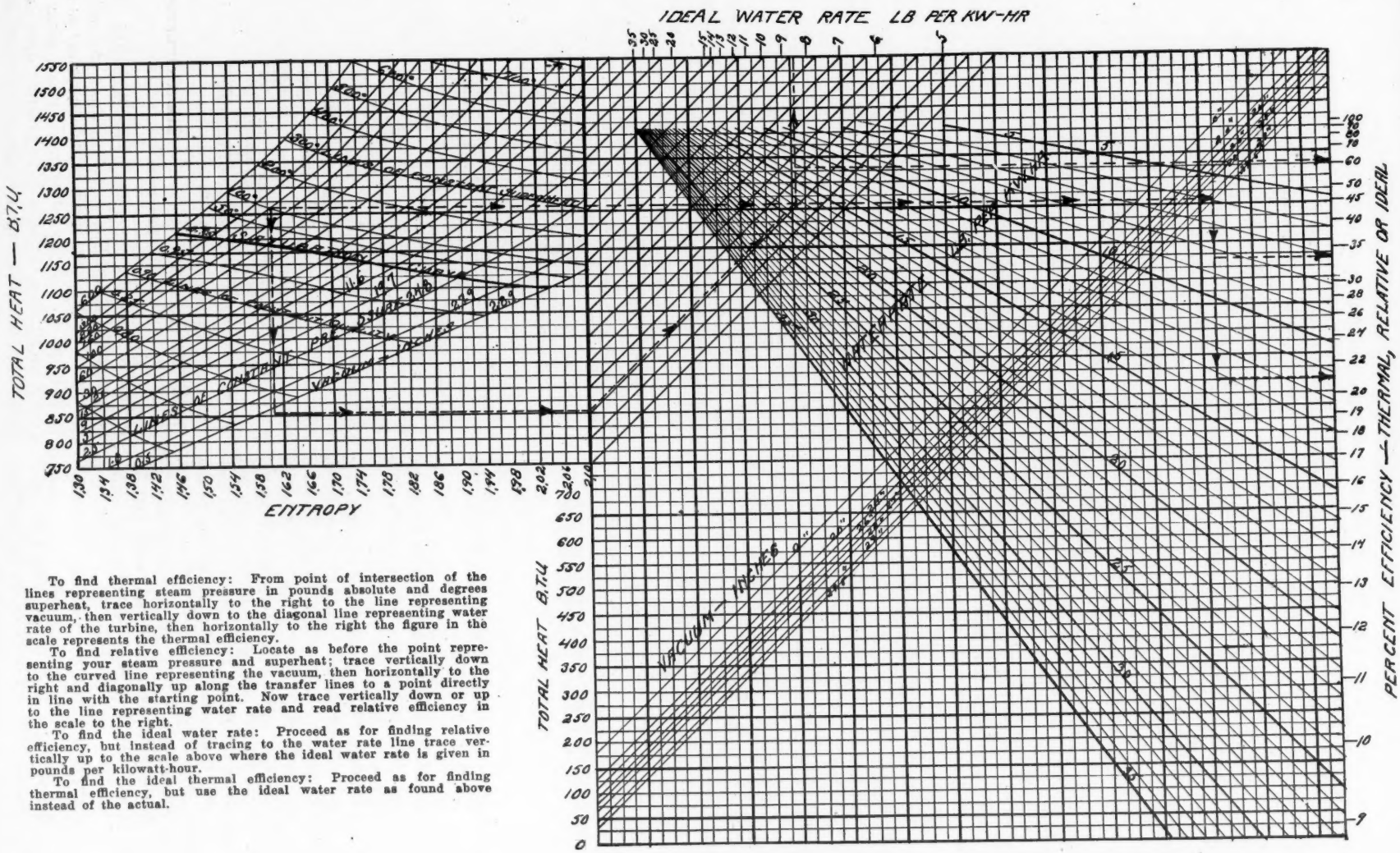
By relative efficiency is meant the ratio of actual thermal efficiency to the ideal efficiency. Knowing these efficiencies, there is a basis from which to work and an ideal for which to strive.

From the above it will be readily seen that any condition which will increase or decrease the value of $(H_1 - H_2) \div (H_1 - q_2)$ will increase or decrease the theoretical efficiency. Thus increasing H_1 will increase the efficiency; this may be done by increasing the pressure or superheat of the steam at the throttle. Decreasing H_2 will also increase the efficiency; this may be done by lowering the exhaust pressure.

For approximating these efficiencies and studying the effects of varied operating conditions, the accompanying chart, Fig. 1, will be found of value. In the section to the left is the heat entropy diagram. By means of that section to the right, the necessary calculations consisting of substraction, multiplication and division may be performed graphically. While the directions in the corner of the chart explain how it may be used, an analysis showing just how the results are obtained will aid in giving a better understanding of its use.

To obtain thermal efficiency, the first operation is to subtract the heat in the water at exhaust pressure from the total heat of the steam at initial pressure and superheat; this gives $H_1 - q_2$ and is done by passing directly from the point on the heat-entropy chart to the right to the diagonal line representing the vacuum carried in the condenser. This difference could be represented on a scale plotted evenly at the top or bottom of the section to the right; but as it is not necessary to know this intermediate value, the scale has been omitted. These diagonal lines representing vacuums are drawn at an angle of 45 deg. from the horizontal and pass through points in the scale to the left representing, respectively, the heat in the water at the various vacuums.

Having found this difference $(H_1 - q_2)$, it is multiplied by the water rate in pounds per kilowatt-hour as determined by test or otherwise. This is done by passing


IDEAL WATER RATE LB PER KW-HR
TOTAL HEAT — B.T.U.
ENTROPY
TOTAL HEAT B.T.U.
PER CENT EFFICIENCY — THERMAL, RELATIVE OR IDEAL
VACUUM INCHES
Vacuum Pockets
Pounds of Condensing Water per Pound of Steam
Pounds of Steam per Kilowatt-hour

To find thermal efficiency: From point of intersection of the lines representing steam pressure in pounds absolute and degrees superheat, trace horizontally to the right to the line representing vacuum, then vertically down to the diagonal line representing water rate of the turbine, then horizontally to the right the figure in the scale represents the thermal efficiency.

To find relative efficiency: Locate as before the point representing your steam pressure and superheat; trace vertically down to the curved line representing the vacuum, then horizontally to the right and diagonally up along the transfer lines to a point directly in line with the starting point. Now trace vertically down or up to the line representing water rate and read relative efficiency in the scale to the right.

To find the ideal water rate: Proceed as for finding relative efficiency, but instead of tracing to the water rate line trace vertically up to the scale above where the ideal water rate is given in pounds per kilowatt-hour.

To find the ideal thermal efficiency: Proceed as for finding thermal efficiency, but use the ideal water rate as found above instead of the actual.

FIG. 1. EFFICIENCY AND WATER RATE CHART FOR TURBINES

vertically from the point of intersection of the heat line and the vacuum line to the line radiating from the upper left-hand corner representing water rate, then passing horizontally to the right. This product might be found in an evenly divided scale which has been omitted; instead a scale has been plotted representing this value, $(H_1—q_2)$ W R, divided into 3412, which is the equivalent of 1 kw.-hr. This gives the thermal efficiency, the result desired, and solves the equation $3412 \div (H_1—q_2)$ W R.

To get the relative efficiency we divide the thermal efficiency by the ideal efficiency, or

$$\frac{3412}{(H_1—q_2)\ W\ R} \div \frac{H_1—H_2}{H_1—q_2} = \frac{3412}{(H_1—H_2)\ W\ R}$$

H_2 is found on the chart by passing vertically down on the entropy chart from the point representing initial steam pressure to the line representing vacuum in the condenser and reading the scale to the left. Instead of passing to the left, however, pass horizontally to the right and along the diagonal transfer line to the heat line representing H_1. An even scale at the top, not

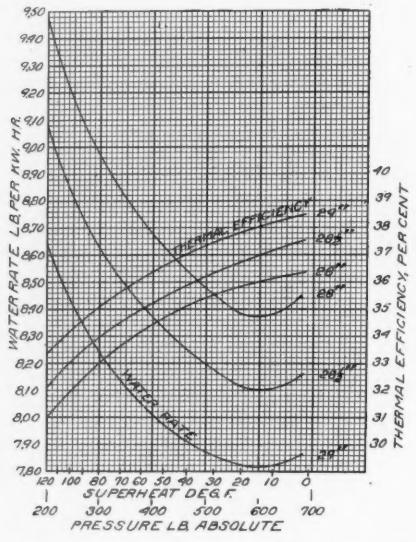


FIG. 2. THERMAL EFFICIENCIES AND WATER RATES WITH STEAM AT 500 DEG. F.

given, would give $H_1—H_2$. This multiplied by the water rate and dividing the product into 3412, as before, gives the relative efficiency in the scale to the right.

Now, if the relative efficiency were 100 per cent, the water rate must be the ideal since we are assuming ideal, *i. e.*, adiabatic expansion of the steam. To get the ideal water rate, then, instead of multiplying by the actual rate, as in finding the relative efficiency, pass up to the point opposite the 100 per cent point on the scale, and read the ideal water rate on the radiating lines. For convenience, however, this scale is placed at the top of the chart as indicated.

The ideal efficiency is found in the same way as the thermal efficiency except that the ideal water rate, as determined above, is used instead of the actual water rate.

Effects of Pressure Changes

In the past few years considerable thought has been given to the advantages to be gained by increasing steam pressure to 500 or 600 lb. from the usual practice of 200 to 250 lb. in modern central stations. While the discussions so far have been principally academic, the advantages to be gained seem to warrant practical application which, according to manufacturers of boilers and turbines, is entirely feasible. The accompanying curves, Fig. 2, from a recent issue of Electrical World, show the decrease in water rate and increase in efficiency which

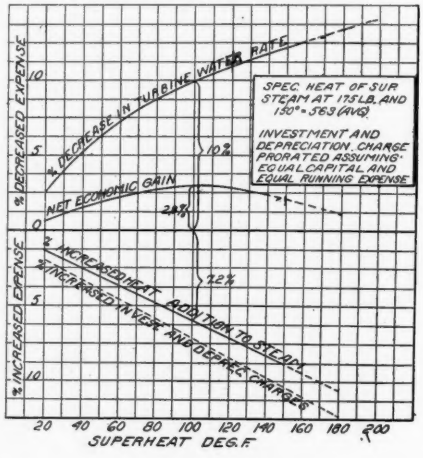

% DECREASED EXPENSE
% DECREASE IN TURBINE WATER RATE
SPEC. HEAT OF SUR STEAM AT 175 LB. AND 150°=.563 (AVG.)
INVESTMENT AND DEPRECIATION CHARGE PRORATED ASSUMING EQUAL CAPITAL AND EQUAL RUNNING EXPENSE
10%
NET ECONOMIC GAIN
2.8%
7.2%
% INCREASED EXPENSE
% INCREASED DIRECT ADDITION TO STEAM
% INCREASED INVEST. AND DEPRECIATION CHARGES
SUPERHEAT DEG. F.

FIG. 3. NET IMPROVEMENT IN PLANT EFFICIENCY WITH VARYING DEGREES OF SUPERHEAT

are theoretically obtained by increasing the pressure of the steam, but maintaining the temperature constant at 500 deg. F. by decreasing the superheat.

Advantages of Superheat

Turbine construction is well adapted to the use of steam of high temperature as there are no rubbing parts

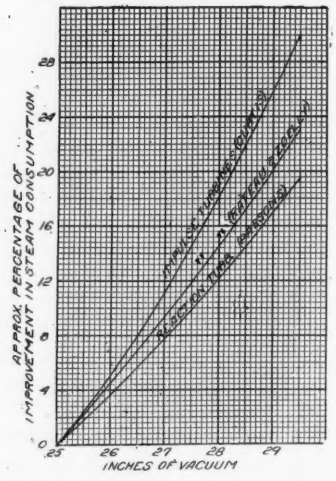


FIG. 4. INFLUENCE OF VACUUM ON STEAM CONSUMPTION OF TURBINES

to become bound by expansion of the materials. Temperatures of 650 deg. F. are not uncommon and turbines have been designed for steam with temperature as high as 800 deg. F. Highly superheated steam gives best results in turbines having a large pressure drop in the first stage, in which case the high temperature steam

comes in contact only with the steam passages and the first stage nozzles. Single-stage impulse type turbines are best suited for high temperature steam, while multi-stage reaction types are least suited. This is one of the factors which has led to the use of impulse blading for the first stage with intermediate and low-pressure stages of the reaction type. Besides the increase in efficiency from a thermodynamic standpoint, by the use of superheat there is the added advantage of less erosion of the blades due to freedom from moisture in the steam and a decrease in rotation losses. It is roughly estimated that there is a gain of one per cent for each 7 to 13 deg. F. of superheat in high-pressure turbines, the better value being for superheat up to 50 deg. F., while the other holds up to 200 deg. Superheating shows greater gains with non-condensing than with condensing turbines. The most economical degree of superheat, as shown in the accompanying curves, Fig. 3, given by the Westinghouse Elec. & Mfg. Co., is 100 deg. F. While the water rate

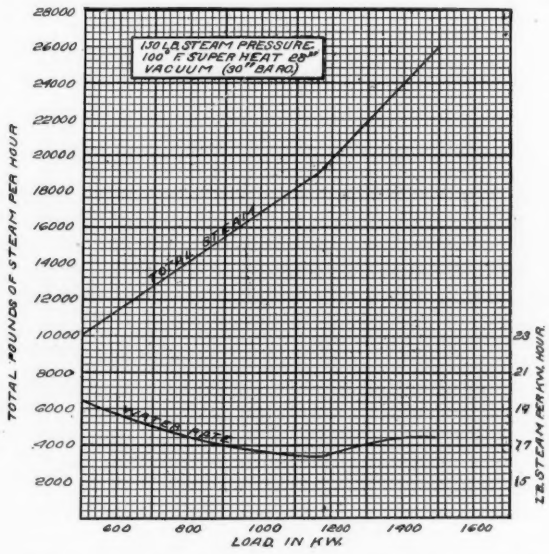


FIG. 5. WATER RATE CURVE FOR 1000-KW. TURBINE

decreases with higher temperatures, the added expense in raising the temperature offsets the advantages.

Gain in efficiency due to high vacuum is very marked particularly in low-pressure turbines properly designed to take advantage of high vacuum conditions. Turbines can use to advantage vacuums as high as 29.5 in. based on a 30-in. barometer. The power required to produce this vacuum under favorable cooling water conditions ranges from 1.5 to 3 per cent of the turbine output for large units.

No general statement can be made in regard to the effect of vacuum on steam consumption, as this varies greatly with the type of turbine. For rough estimation, Lowenstein states that a decrease in steam consumption of 5 per cent for each inch increase in vacuum between 24 to 27 in. is reasonable for high-pressure turbines, 6 to 7 per cent between 27 and 28 in., and 7 to 8 per cent between 28 and 29 in. The gain in low-pressure turbines due to increase in vacuum is greater, but the limit is reached when the velocity through the buckets reaches the acoustic point (about 1400 ft. per sec.). Further increase in vacuum has no effect on steam economy and to secure higher efficiency would require increased area through the blades; the usual method, however, is to add more stages with increased area.

The water rate curve for a given turbine is nearly a straight line when total flow of steam is plotted against load.

The steam used increases from zero to rating, at which point the flow begins to increase more rapidly than the load, giving a slight bend in the curve. Figure 5 gives a general idea of this relation. There are, however, variations from this line giving increased steam consumption at those loads where an additional nozzle has just been opened.

Turbines of large size are decidedly more economical of steam than those of small rating, as is shown by the accompanying table compiled by Loewenstein. This fact is partly due to the fact that the smaller units are frequently used as auxiliaries or for special industrial purpose where the heat of the exhaust steam is employed for heating and economy of steam is not of highest importance.

WATER RATES OF TURBO-GENERATORS

(60-cycle, 2300-volt a.c. generators)

Condensing				Non-condensing	
Steam at 150 lb. gage, 28-in. vacuum		Steam at 200 lb. gage, 150 deg. superheat, 29-in. vacuum		Steam at 150 lb. gage Exhaust to Atmosphere	
Rating kw.	Water rate, lb. per hr. per kw. incl. Excitation	Rating kw.	Water rate, lb. per hr. per kw. incl. Excitation	Rating kw.	Water rate, lb. per hr. per kw.
50	32.0 to 42.0	4,000	12.0 to 15.3	50	52.0 to 60
100	21.2 to 27.5	5,000	11.9 to 13.0	100	43.5 to 50
200	19.5 to 25.0	7,500	11.6 to 12.6	200	38.4 to 44
300	18.7 to 22.5	10,000	11.2 to 12.3	300	36.5 to 42
400	18.2 to 21.0	12,500	11.1 to 12.0	400	35.2 to 40
500	17.7 to 19.3	15,000	11.1 to 11.8	500	34.1 to 38
600	17.4 to 19.0	17,500	11.0 to 11.8	600	33.4 to 37
750	17.0 to 18.8	20,000	11.0 to 11.4	750	32.7 to 36
1,000	16.5 to 18.6	25,000	10.9 to 11.3	1,000	31.7 to 35
1,250	16.2 to 18.3	30,000	10.8 to 11.2	1,250	31.0 to 34
1,500	16.0 to 18.0	35,000	10.7 to 11.1	1,500	30.7 to 33
2,000	15.7 to 17.7	40,000	10.6 to 11.1	2,000	30.1 to 32
2,500	15.5 to 17.4	50,000	10.5 to 11.0	2,500	29.4 to 31
3,000	15.4 to 17.1	60,000	10.5 to 10.9	3,000	28.9 to 30
3,500	15.3 to 16.9	75,000	10.4 to 10.8	3,500	28.6 to 29

*Speeds: 50 to 3500 kw., inc., 3600 r.p.m.; 4000 to 20,000 kw., inc., 1800 r.p.m.; 25,000 kw. and over, 1200 r.p.m.

THE SUITABILITY of any particular fuel may be determined by its tendency to cause smoke emission or to produce objectionable clinkers under some conditions, or by other factors. Frequently, however, a fuel considered unsatisfactory by the operating force in a given plant may, when differently handled, prove to be suitable. This statement applies to power plants or heating plants of considerable size, but is just as true when applied to residence heating conditions and equipment.—Bureau of Mines Yearbook.

MORE ATTENTION than ever before is being paid this winter to careful use of exhaust steam in power plants. It may be applied to innumerable purposes, such as heating feed water for the steam boiler, for many washing purposes, heating buildings, pasteurizing and sterilizing, and the like. A small investment in additional boiler-room equipment, such as an exhaust-steam heater, will effect savings of several hundred dollars a year in the coal bill of even a moderate-sized power plant.

Condensers

EXAMPLES SHOWING BENEFITS OF INCREASED VACUUM; CONDENSER INSTALLATIONS

IN ANY steam engine or turbine in which the expansion of the steam is adiabatic, the energy convertible into work is directly proportional to the difference between the heat content of the steam at the beginning and that at the end of expansion; therefore, the greater the number of expansions occurring the greater the number of heat units per pound of steam available for useful work and the higher the efficiency of the machine. With a turbine receiving steam at an absolute pressure of 160 lb. per square inch and exhausting against an atmospheric pressure of 14.7 lb. per square inch, each pound of steam consumed can contribute as much heat only as the difference between the heat values corresponding to these pressures which from the Total Heat-Entropy Chart, page 13, are 1196 and 1022 respectively. The total heat then available for useful purposes is equal to 1196 minus 1022, or 174 B.t.u.

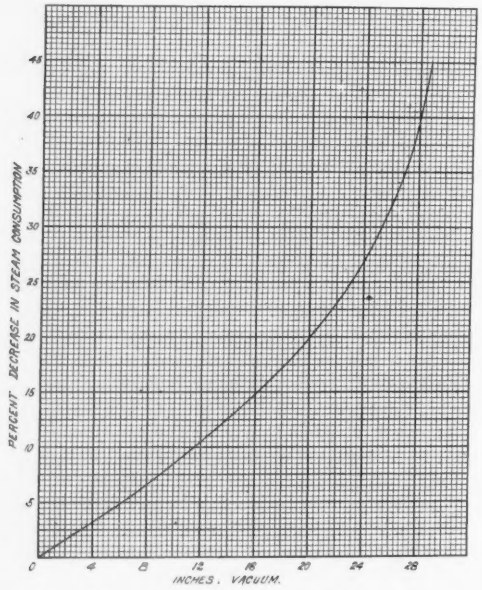


FIG. 1. VARIATION IN STEAM CONSUMPTION DUE TO USE OF CONDENSER

If, however, a condenser is added and the back pressure reduced to 2 lb. absolute, corresponding to a 26-in. vacuum, the total heat then available is equal to 1196 minus 907, or 289 B.t.u., or a gain of 115 B.t.u. for each pound of steam. With further pressure reduction, let us say to 0.696 lb. per square inch absolute (28.6-in. vacuum), 1196 minus 860, or 336 B.t.u., are available, 162 more than with the engine or turbine exhausting against atmospheric pressure or 47 B.t.u. more than when operating under a 26-in. vacuum. The per cent gain in useful work with 26 and 28.5 in. of vacuum is 66 and 93 respectively; and with a 28.5-in. vacuum, 16 per cent more heat may be usefully employed than is possible with a 26-in. vacuum.

One horsepower-hour is equivalent to 2545 B.t.u., thus giving a theoretical steam consumption of 2545 divided by 289, or 8.8 lb. per horsepower-hour, when operating under a 26-in. vacuum, and 2545 divided by 336, or 7.5 lb. per horsepower-hour with a 28.5-in. vacuum, or a theoretical reduction in steam consumption, by the use of a 28.5-in. vacuum as compared with a 26-in. vacuum, of nearly 15 per cent.

The advantages accruing from the use of condensers in conjunction with steam turbines and especially when high vacuums are employed, are thus readily apparent.

By the above methods of calculation, we find that with a turbine operating under an initial pressure of 200 lb. per square inch absolute, and vacuum of 26-in., the theoretical steam consumption in pounds per horsepower-hour is equal to 8.74, while with an initial pressure of 50 lb. absolute per square inch and the same degree of vacuum, the theoretical steam consumption increases to 9.54 lb. per horsepower-hour. If, however, we

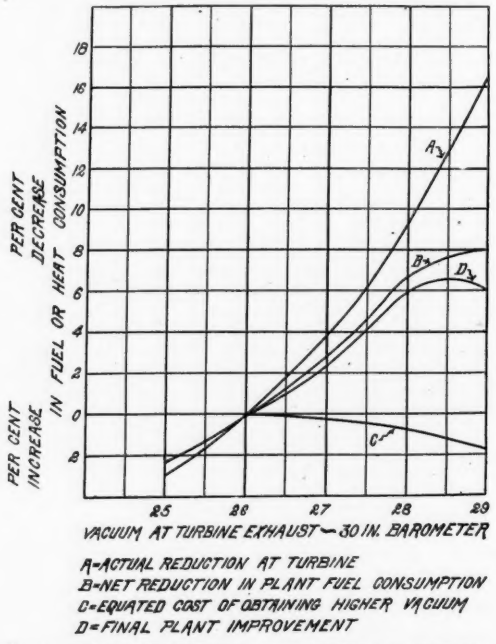


FIG. 2. ACTUAL INFLUENCE OF VACUUM ON POWER OF A 200-KW. TURBINE

employ a lower vacuum of 28.5 in., the steam consumption with the same pressures of 200 and 50 lb. per square inch absolute becomes 7.52 and 7.11 lb. per horsepower-hour, or a reduction in steam consumption of 13.9 per cent when initial pressure is 200 lb. per square inch absolute as against a 15 per cent reduction with an initial pressure of 50 lb., thus showing that the per cent gain in running condensing is greater with low-pressure than with high-pressure steam. It is evident also that the gain realized from the last few inches of vacuum is relatively much more than from the first few inches.

From a practical consideration of the subject, however, due to the increasing cost of elaborate condensing equipment required for the production of high vacuums and the varying characteristics of the different types of turbines, the effect of vacuum on steam consumption is not quite as pronounced as may appear from the preceding discussion. In general, for high-pressure units there

is a decrease in steam consumption of approximately 5 per cent for each inch of vacuum secured between 24 and 27 in., and of 7 to 8 per cent between 28 and 29 in. And, as stated before, the gain due to the use of vacuums in connection with low-pressure units is even greater than this. The variation in steam consumption which may be expected due to operating condensing is indicated in Fig. 1. In fact, on account of the rapidly increasing volume of steam with the decrease in back pressure, due to the vacuum, the corresponding circulating and air pump capacity and power demand become proportionately larger and as a consequence a point is reached where the improvement in steam economy fails to exceed the increased power demanded by the auxiliaries. This is well illustrated in Fig. 2, presented in a discussion before the A. S. M. E. on the paper of S. L. Naphtaly, entitled, "Test of a 10,000-Kw. Steam Turbine," which indicates the influence of vacuum on power. The turbine, the test results of which were used in the plotting of these curves, was a 200-kw. unit using steam at 175 lb. pressure and 100 deg. F., superheat, and equipped with a surface condenser supplied with injection water at an average temperature of 55 deg. F.

Condenser Details

Due to the fact that they should be placed in close proximity to the turbine, surface and some forms of jet condensers are preferable for use in conjunction with this type of prime mover. The former are especially applicable for this service when the turbine is to operate against extremely high vacuums, and wherever the cooling water is not fit for boiler feed or no suitable and cheap supply of pure boiler feed is available.

Practically the cooling surface of any surface condenser is calculated as tube surface only. The number of square feet of tube surface required depends on three factors: Weight of steam required to be condensed per hour; total heat above 32 deg. F., of the exhaust steam at condenser pressure minus the heat in the condensate at the final temperature of the condensed steam; and B.t.u. transmitted per hour per square foot of cooling surface for each degree difference between the temperature of the exhaust steam at condenser pressure and the mean temperature of the circulating water. This last expression is known as the coefficient of heat transmission. Since the heat given up by the steam is equal to that absorbed by the cooling water, the product of the number of square feet of cooling surface times the coefficient of heat transmission times the difference in mean temperature is equal to the weight of condensed steam per hour times (the total heat of the exhaust steam at condenser pressure less the final temperature of the condensed steam plus 32). Hence, to find the cooling surface in square feet, divide (weight of condensed steam per hour times the difference between total heat of the exhaust steam at condenser pressure and the final temperature of the condensed steam plus 32) by (the coefficient of heat transmission times the difference between exhaust steam temperature at condenser pressure and the mean circulating water temperature.*

Table I, reproduced from Mark's Mechanical Engineers' Handbook, gives the amount of cooling surface for various values of temperature rise in the water. It is based on a value of 350 for U, the coefficient of heat transmission, an average taken from a large number of installations. For other values of U, multiply the figure taken from the table by 350 and divide by the new value of U to obtain the necessary surface.

TABLE I. SQUARE FEET CONDENSING SURFACE PER 1000 LB. STEAM FOR VARIOUS VALUES OF TEMPERATURE RISE IN COOLING WATER

Vacuum in. 30 in. bar.	Temp. rise in water deg. F.	Inlet Water Temperature, Degrees Fahrenheit.						
		50	55	60	65	70	75	80
29	5	107.1	132.3	172.0	249.0			
	10	119.6	152.3					
	15	137.4	184.2					
	20	165.0						
28.5	5	72.9	83.3	98.0	117.5	149.0	202.0	
	10	78.1	90.5	107.9	133.3	175.3		
	15	84.5	99.7	121.5	156.0			
	20	93.2	112.0	141.5	196.0			
	25	104.2	130.0	176.4				
	30	120.5	161.0					
28	5	58.6	65.2	73.7	84.9	99.5	121.0	153.7
	10	61.8	69.7	79.5	92.3	110.3	137.0	182.3
	15	66.1	74.6	86.4	102.0	124.8	161.8	
	20	70.7	81.2	95.2	113.2	146.0		
	25	76.6	89.0	107.0	134.0	184.7		
	30	84.0	99.9	124.0	168.0			
	40	108.8	143.6					
27.5	5	50.3	55.8	61.6	69.4	79.0	91.5	108.2
	10	53.5	59.4	65.8	74.0	85.2	100.1	121.5
	15	56.1	62.3	69.9	80.0	93.2	111.9	140.1
	20	59.5	66.5	75.4	87.3	103.5	128.2	169.4
	25	63.4	71.4	82.0	96.6	118.2	153.7	
	30	68.0	77.8	91.0	110.0	141.3		
	40	81.6	97.3	122.2	174.6			
27	5	45.7	49.4	54.0	59.8	67.2	76.4	86.5
	10	47.4	51.8	57.2	63.6	71.5	82.2	96.1
	15	50.0	54.5	60.6	67.8	77.0	89.6	106.2
	20	52.7	57.8	64.3	72.8	83.6	98.7	120.4
	25	55.4	61.8	69.0	79.0	92.4	111.6	142.3
	30	58.9	66.8	75.0	87.0	104.5	131.3	184.0
	40	66.2	78.2	92.5	114.3	155.7		
26.5	5	41.8	45.4	49.3	53.6	59.2	66.7	74.6
	10	43.5	47.4	51.4	56.5	62.8	70.7	80.6
	15	45.5	49.5	54.2	59.8	67.1	75.8	87.8
	20	47.6	52.0	57.2	63.6	71.7	82.5	97.0
	25	50.0	54.8	61.1	68.2	77.9	90.8	109.4
	30	52.7	58.2	65.2	73.9	85.6	102.2	128.0
	40	59.7	67.3	75.6	90.9	111.7	150.0	
26	5	39.6	42.1	45.4	49.4	54.0	59.2	66.8
	10	40.8	43.4	47.4	51.3	56.9	63.2	71.1
	15	42.1	45.5	49.5	54.2	60.0	67.1	76.3
	20	43.9	47.7	52.1	57.4	63.8	72.1	82.7
	25	45.9	50.0	55.0	60.9	68.5	78.1	91.1
	30	48.3	52.9	58.4	65.2	73.9	86.0	102.7
	40	54.0	59.8	67.5	77.2	91.2	112.5	161.0
25	5	34.8	37.3	40.2	43.1	46.5	50.9	55.3
	10	36.0	38.7	41.5	44.8	48.6	53.3	58.6
	15	37.4	40.2	43.2	46.8	50.8	56.2	62.2
	20	38.95	41.9	45.1	48.9	53.6	59.4	66.5
	25	40.55	43.6	47.3	51.5	56.7	63.3	71.3
	30	42.2	45.5	49.6	54.5	60.5	68.0	77.6
	40	46.3	50.6	55.7	62.2	70.5	81.5	97.2

Surface condensers for large turbine installations have generally from 1¼ to 2 sq. ft. of condensing surface per kilowatt, or approximately 1/12 to 1/7 sq. ft. for each pound of steam condensed. Small turbines require from 2 to 4 sq. ft. per kilowatt or from 1/10 to 1/5 sq. ft. per pound of steam.

On account of its better heat transmission, condenser tubes are generally made of brass or other composition containing a greater or less percentage of copper with diameters ranging from ½ to 1 in. These sizes appear to represent established practice, as smaller tubes introduce excessive resistance to the flow of water and larger ones, in order to be of sufficient strength to carry their own weight and that of the contained water, require a

*Let S be the cooling surface in square feet; Hs the total heat of the exhaust steam at condenser pressure, and Ts its temperature; T_1 the initial and T_2 the final temperature of circulating water; To the final temperature of condensed steam; all in deg. F.; U the coefficient of heat transmission; d the difference in mean temperatures, which according to Whitham is:

$$d = Ts - \frac{T_1 + T_2}{2}$$

and W the weight of condensed steam per hour.

We have $SUd = W\,[Hs - (To - 32)]$

$$S = \frac{W\,[Hs - (To - 32)]}{Ud}$$

With circulating water velocities of from 25 to 50 ft. a minute, the value of U with plain brass tubes is 250, and for corrugated tubes 300; when the velocities vary from 75 to 150 ft. per minute, the above values of U may be increased by 50 per cent.—Steam Power Plant Engineering—Gebhardt.

thickness resulting in reduction of heat transmission and efficiency. Sometimes for protection against corrosion the tubes are tinned on the inside or on both sides.

The shell must be of such size as to provide for ready inspection, cleaning and repair with heads of sufficient volume to care for a continuous and unrestricted flow of cooling water. The shell is designed and the tubes arranged so that the upper portion has a volume equal to approximately from ¼ to ⅓ of the shell unoccupied; the steam is compelled to pass between all of the tubes and provision is made in the lower compartment for cooling the air and water before discharge.

While there are no fixed rules or formulas governing the size of jet condenser chambers required for various services, they must not be so small as to result in continual flooding and its detrimental results to the unit it is serving, nor so large as to require too great a length of time for the air pump to create a vacuum.

TABLE II. NUMBER OF POUNDS OF COOLING WATER REQUIRED PER POUND OF STEAM IN SURFACE CONDENSERS

Temp. of Cooling Water, Deg. F.	Temp. of Discharge Water, Deg. F.	Vacuum, Inches Mercury						
		22	24	26	27	28	29	29.5
35	80	22.3	22.5	22.6	22.8	22.9	23.2	23.5
	90	18.3	18.3	18.4	18.5	18.8	19.0	19.2
	100	15.4	15.5	15.6	15.8	15.9	16.1	16.2
	110	13.5	13.6	13.6	13.6	13.7	13.9	14.1
	120	11.8	11.9	12.0	12.0	12.1	12.3	12.4
	130	10.6	10.6	10.7	10.8	10.8	11.0	11.1
	140	9.5	9.6	9.7	9.8	9.8	9.9	10.0
40	80	25.1	25.3	25.5	25.6	25.8	26.1	26.4
	90	20.1	20.2	20.4	20.5	20.6	20.9	21.1
	100	16.7	16.8	17.0	17.1	17.2	17.4	17.6
	110	14.3	14.4	14.5	14.6	14.7	14.9	15.1
	120	12.5	12.6	12.7	12.8	12.9	13.0	13.2
	130	11.1	11.2	11.3	11.4	11.4	11.6	11.7
	140	10.0	10.1	10.2	10.2	10.3	10.4	10.5
50	80	33.5	33.7	34.0	34.2	34.4	34.9	35.2
	90	25.1	25.3	25.5	25.6	25.8	26.1	26.4
	100	20.1	20.2	20.4	20.5	20.6	20.9	21.1
	110	16.7	16.8	17.0	17.1	17.2	17.4	17.6
	120	14.3	14.4	14.5	14.6	14.7	14.9	15.1
	130	12.5	12.6	12.7	12.8	12.9	13.0	13.2
	140	11.1	11.2	11.3	11.4	11.4	11.6	11.7
60	80	50.3	50.6	51.0	51.3	51.7	52.3	52.9
	90	33.5	33.7	34.0	34.2	34.4	34.9	35.2
	100	25.1	25.3	25.5	25.6	25.8	26.1	26.4
	110	20.1	20.2	20.4	20.5	20.6	20.9	21.1
	120	16.7	16.8	17.0	17.1	17.2	17.4	17.6
	130	14.3	14.4	14.5	14.6	14.7	14.9	15.1
	140	12.5	12.6	12.7	12.8	12.9	13.0	13.2
70	80	100.6	101.2	102.1	102.7	103.4	104.7	105.8
	90	50.3	50.6	51.0	51.3	51.7	52.3	52.9
	100	33.5	33.7	34.0	34.2	34.4	34.9	35.2
	110	25.1	25.3	25.5	25.6	25.8	26.1	26.4
	120	20.1	20.2	20.4	20.5	20.6	20.9	21.1
	130	16.7	16.8	17.0	17.1	17.2	17.4	17.6
	140	14.3	14.4	14.5	14.6	14.7	14.9	15.1
80	90	100.6	101.2	102.1	102.7	103.4	104.7	105.8
	100	50.3	50.6	51.0	51.3	51.7	52.3	52.9
	110	33.5	33.7	34.0	34.2	34.4	34.9	35.2
	120	25.1	25.3	25.5	25.6	25.8	26.1	26.4
	130	20.1	20.2	20.4	20.5	20.6	20.9	21.1
	140	16.7	16.8	17.0	17.1	17.2	17.4	17.6
90	100	100.6	101.2	102.1	102.7	103.4	104.7	105.8
	110	50.3	50.6	51.0	51.3	51.7	52.3	52.9
	120	33.5	33.7	34.0	34.2	34.4	34.9	35.2
	130	25.1	25.3	25.5	25.6	25.8	26.1	26.4
	140	20.1	20.2	20.4	20.5	20.6	20.9	21.1

The required area of the steam inlet is governed by the diameter of the pipe* required to carry the steam to the condenser, found by multiplying 0.175 by (the square root of the quotient from dividing the weight of steam flowing in pounds per minute, by the density of the steam in pounds per cubic foot).

Tables II and III indicate the number of pounds of cooling water required per pound of steam in surface condensers and the number of pounds injection water required per pound of steam in jet condensers, respectively. From these values the size of water inlet is governed by the size of pipe required to conduct the water from the source of supply to the condenser at a reasonable loss from friction.

*The weight of steam flowing in pounds per minute is equal to the maximum steam consumption of the turbine used in connection with the condenser. The value of density is obtained from a steam table for the average pressure to be carried in the condenser.

Injection pipe diameters are generally based on a water velocity ranging from 400 to 600 ft. per min., and discharge pipe diameters on velocities of from 200 to 400 ft. per min.

High vacuum systems used in connection with steam turbines, where condenser pressures of ½ lb. absolute are maintained, call for refinements not found in ordinary condensing systems. In addition to the circulating or injection water pumps, air pumps are employed for the removal of air and noncondensible vapors, sometimes augmented by air coolers built inside or outside of the condenser for the purpose of cooling the air and removing any water that may remain.

If a reciprocating dry vacuum pump is employed, the circulating pump requires 85 to 90 per cent of the total horsepower for condenser auxiliaries; the condensate and dry vacuum pumps 5 to 7 per cent each. The total horsepower required at 27 to 28 in. vacuum is

TABLE III. NUMBER OF POUNDS INJECTION WATER REQUIRED PER POUND OF STEAM IN JET CONDENSERS

Hot Well Temp. Deg.F.	Initial Temp. Injection Water Deg. F.	Vacuum, Inches Mercury						
		22	24	26	27	28	29	29.5
120	35	12.2	12.1	12.0	11.9	11.9	11.8	11.6
	40	12.9	12.8	12.7	12.7	12.6	12.5	12.4
	50	14.8	14.7	14.6	14.5	14.4	14.3	14.1
	60	17.3	17.1	17.0	16.9	16.8	16.7	16.5
	70	20.7	20.6	20.4	20.3	20.2	20.1	19.8
	80	25.9	25.6	25.5	25.4	25.3	25.0	24.8
	90	34.6	34.3	34.0	33.9	33.7	33.4	33.1
	100	51.8	51.5	51.1	50.9	50.6	50.1	49.6
	110	103.5	103.0	102.2	101.8	101.2	100.2	99.3
110	35	13.9	13.8	13.7	13.7	13.6	13.5	13.2
	40	14.9	14.8	14.7	14.6	14.6	14.4	14.3
	50	17.4	17.3	17.2	17.1	17.0	16.8	16.7
	60	20.9	20.8	20.6	20.5	20.4	20.2	20.0
	70	26.1	26.0	25.8	25.7	25.5	25.3	25.0
	80	34.8	34.6	34.4	34.2	34.1	33.7	33.4
	90	52.2	52.0	51.6	51.4	51.1	50.6	50.1
	100	104.5	104.0	103.2	102.8	102.3	101.2	100.3
100	35	16.2	16.1	16.0	15.9	15.8	15.7	15.5
	40	17.5	17.5	17.3	17.3	17.2	17.0	16.9
	50	21.1	21.0	20.8	20.7	20.6	20.4	20.2
	60	26.3	26.2	26.0	25.9	25.8	25.5	25.3
	70	35.1	35.0	34.7	34.6	34.4	34.0	33.7
	80	52.7	52.5	52.1	51.9	51.6	51.1	50.6
	90	105.5	105.0	104.2	103.8	103.3	102.2	101.3
90	35	19.3	19.2	19.0	19.0	18.9	18.7	18.6
	40	21.3	21.2	21.0	20.9	20.8	20.6	20.4
	50	26.6	26.5	26.3	26.2	26.0	25.8	25.5
	60	35.5	35.3	35.0	34.9	34.7	34.4	34.1
	70	53.2	53.0	52.6	52.4	52.1	51.6	51.1
	80	106.5	106.0	105.2	104.8	104.3	103.2	102.3
80	35	23.9	23.7	23.6	23.5	23.4	23.1	22.9
	40	26.8	26.7	26.5	26.4	26.3	26.0	25.7
	50	35.8	35.6	35.4	35.2	35.1	34.7	34.4
	60	53.7	53.5	53.1	52.9	52.6	52.1	51.6
	70	107.5	107.0	106.2	105.8	105.3	104.2	103.3
70	35	31.0	30.8	30.6	30.5	30.3	30.0	29.8
	40	36.1	36.0	35.7	35.6	35.4	35.0	34.7
	50	54.2	54.0	53.6	53.4	53.1	52.6	52.1
	60	108.5	108.0	107.2	106.8	106.3	105.2	104.3

about 2 to 2.5 per cent that of the main unit; for vacuums of from 28 to 29 in., this is increased to 3 to 3.5 per cent.

Arrangement of Condensing Apparatus

Independent condensing systems in which each unit is provided with its individual condenser and auxiliaries, are now employed almost exclusively except in the case of the smaller sized turbines. The condenser should be placed below and as near the turbine as possible, in order that the exhaust steam may gravitate toward the condenser, and in exhaust piping the number of bends and joints should be reduced to a minimum to decrease friction and the possibility of air leakage; siphon and barometric condensers, due to their nature, cannot be so placed.

On account of local conditions, it may be found necessary to place jet or surface condensers on the same level or above the turbine, requiring the use of addi-

tional piping and bends. In this case, the lowest point of the exhaust piping should be fitted with a drain which should be open whenever the turbine is not operating.

In high-vacuum systems the condenser should always be placed directly below the turbine, the steam connections being joined by a short copper corrugated expansion joint.

Centrifugal condensers, like the ordinary jet or surface, are preferably connected as directly and closely as possible to the turbine, and are generally mounted also

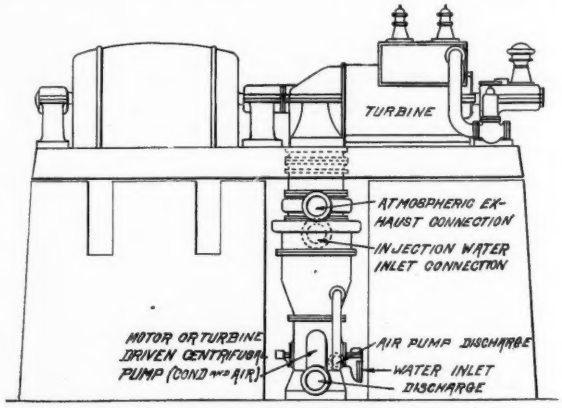


FIG. 3. TYPICAL TURBINE INSTALLATION EMPLOYING A JET CONDENSER

on the suction of the wet vacuum pump which in some instances is built integral with the condenser.

The height of a barometric or jet condenser above the source of injection or cooling water supply should not exceed 18 or 20 ft. For high lifts, such as in the case of barometric condensers, a charging valve and overhead supply will be found advantageous in starting, this, however, to be shut off as soon as the vacuum has been established.

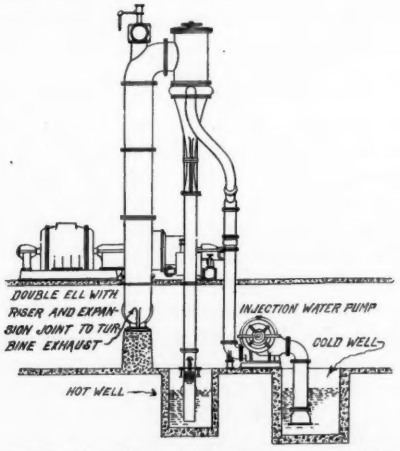


FIG. 4. TYPICAL TURBINE INSTALLATION EMPLOYING A BAROMETRIC CONDENSER

Typical turbine installations employing the various types of condensers are illustrated in Figs. 3 to 5 inclusive.

No exact data is available relative to the amount of space required by condensing equipment, due to the fact that the desired arrangement of piping may be such as to widely separate the various members of the equipment. Generally, however, this is placed within or in close proximity to the foundation housing, and although some engineers object to this on the grounds that being on the floor below the turbine, operators tend to neglect the proper care of the equipment, it is found that with

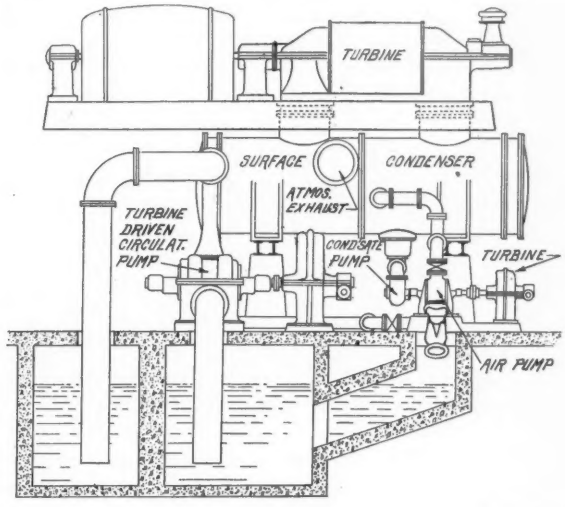


FIG. 5. A HIGH-VACUUM INSTALLATION SHOWING ARRANGEMENT OF SURFACE CONDENSER AND AUXILIARIES

this arrangement a considerable saving in space may be realized.

Steam Regenerators

FOR low-pressure turbines taking steam from reciprocating engines the supply of steam is liable to be quite irregular, varying with the load that the engine has to carry. In order to supply a steady load, from the turbine, or to regulate the power of the turbine to its load, some device must be interposed which stores heat between the engine and turbine during the periods when more steam is available than is needed by the turbine, and can give up heat during the slack periods of steam supply.

Such a device, called a regenerator, has been designed by Rateau, the heat being stored in water, sometimes supplemented by cast-iron plates, the storage being accomplished by a rise in temperature and supply by a corresponding fall.

Two arrangements have been used; a shell set vertical and filled with iron trays over which the water flows in thin sheets, much as in an open type feedwater heater, in order to give intimate contact of steam and water and facilitate transfer of heat.

The horizontal type has a cylindrical shell with baffles to increase circulation, and steam is discharged below the water line through perforations in such a way as to set up rapid currents in the water, as indicated in Fig. 1. No iron plates are used, as the water and steam mix by circulation, and water is the ideal medium for storing and restoring heat energy.

Weight of steam needed per hour by the turbine, temperature variation allowable in the regenerator and the time to run without supply of steam to the generator are the factors which determine the weight of water

needed, so far as turbine operation is concerned. But the limit of heat storage may be set by the maximum amount of steam that has to be taken care of from the engine exhaust while the turbine is running light.

Generally the engine exhaust is taken in at about atmospheric pressure, though sometimes the engine is exhausted at a slight vacuum. The maximum variation of pressure allowable in the regenerator is 2 to 3 lb.

If T be the temperature range allowed, t the time in minutes that the supply may be shut off, and S the steam per hour needed by the turbine, then the heat per minute for the turbine will be $S \times 966 \div 60$, or for t minutes, $S \times 966 \times t \div 60$. The weight of water, W, needed in the regenerator will be:

$$W = S \times 966 \times t \div (60 \times T)$$
$$= S \times t \times 16.1 \div T \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

As the specific heat of cast iron is about 1/9 that of water, 9 lb. of cast-iron plate will replace 1 lb. of water for heat storage purposes.

Assuming an average pressure equal to atmospheric in the regenerator and 3 lb. drop, the temperature range, T becomes about 10 deg. and we shall have:

$$W_a = S \times t \times 1.61 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

When engine exhaust is used as the limiting factor, if E is the maximum weight to be taken care of at atmospheric pressure:

$$W_1 = E \times 966 \div T,$$

or for 10 deg. temperature range

$$W_{1a} = E \times 96.6 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Figure 2 from Harding and Willard gives the pounds of water per pound of steam for various operating conditions.

As equipment on the regenerator will be needed, a relief valve to atmosphere to prevent overpressure in case the turbine shuts down; a steam non-return valve on the outlet to the turbine, to prevent backing up into the regenerator when the turbine is supplied with live steam; and a water check valve on the engine exhaust to keep water from backing up from regenerator to engine.

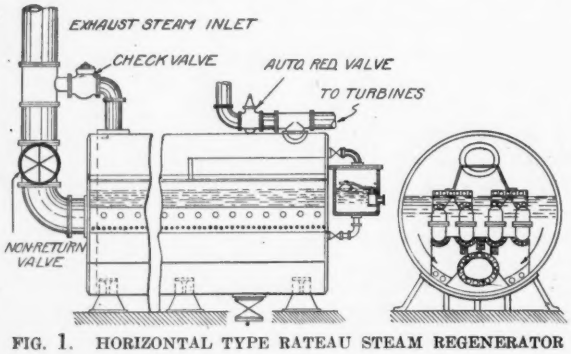


FIG. 1. HORIZONTAL TYPE RATEAU STEAM REGENERATOR

As the water will accumulate slowly in the regenerator due to condensation from radiation and the exhaust steam, it is usual to have a float trap connected to drain off water when it rises above a given level. Also, if it is essential that the turbine be always in operation, an automatic reducing valve must be installed to admit live steam to the turbine in case the engine exhaust is cut off for a period longer than that for which the regenerator can supply steam.

One installation for a 500-kw. turbine has a shell 11.5 ft. in diameter of 3/8-in. steel plate with a central horizontal diaphragm that divides the shell into top and bottom compartments. In each of these are six steam conduits of rectangular section, open at intervals on the bottom and perforated on the sides with many 3/4-in. holes. Water surrounds the conduits and may even submerged them to a depth of 3 in., while steam spaces are connected by vertical pipes. In the upper steam space, a large baffle serves to effect separation of entrained moisture from the steam. The regenerator contains 5.5 tons of water, sufficient to supply the turbine at half load for 7.2 min. or at full load for 6.5 min.

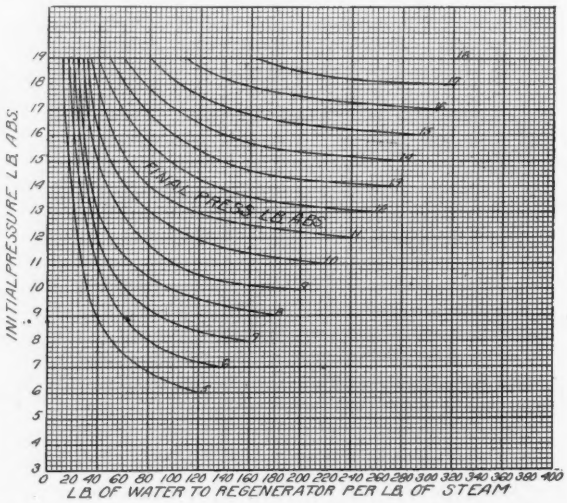


FIG. 2. POUNDS WATER PER POUND OF STEAM FOR REGENERATOR UNDER VARIOUS PRESSURE CONDITIONS

Using Live Steam to Maintain Feed Temperature

DURING the winter months the demands of our heating system are so great that practically all the exhaust steam is used therein, and as a result the feed-water heater is robbed of its supply and a decrease in the feed-water temperature results. Since it has been found extremely desirable to maintain the highest possible feed temperature because of the overload carried on the boilers and the purifying action resulting from the high temperature, it was decided that it would be advantageous to draw some live steam to maintain the feed temperature when sufficient exhaust was not available.

To accomplish this purpose without a waste of steam, a small back-pressure valve was placed on the outlet of the heater so that about 2½ lb. pressure was carried. A 1-in. live steam line was then run to the heater and in it was placed a pressure-reducing valve set to maintain 2½ lb. pressure on its outlet side. As a result, when the exhaust steam began to be insufficient the pressure within the heater would drop below 2½ lb. and then immediately the pressure-reducing valve would open and admit live steam in sufficient quantities to bring the pressure up to the 2½-lb. point. In other words, the pressure-reducing valve thus set acted to keep the heater filled with steam at all times, coming into play when the exhaust steam was insufficient to maintain the heating action.

M. A. SALLER.

Alberger Steam Turbines

TWO general forms of steam turbines are built by the Alberger Pump & Condenser Co. One of these, referred to as the Alberger turbine, is built in capacities up to approximately 40 to 50 hp., while the other, called the Alberger-Curtis steam turbine is manufactured in sizes ranging from 15 to 550 hp. Both are built only for the driving of centrifugal pumps.

In the Alberger turbine, which is of the single-pressure, two-velocity stage type, steam is admitted to the wheel through a divergent nozzle. Leaving the nozzle, the steam enters the passages between the rotor blades where a portion of its velocity energy is extracted and then is directed through a stationary reversing vane which causes it to impinge again upon the rotor buckets when the remaining velocity energy is absorbed.

The bearings, one of which is of the thrust type, are provided with ring oilers and are water jacketed.

Governing is accomplished by means of a throttling flyball governor mounted as indicated in Fig. 1 and

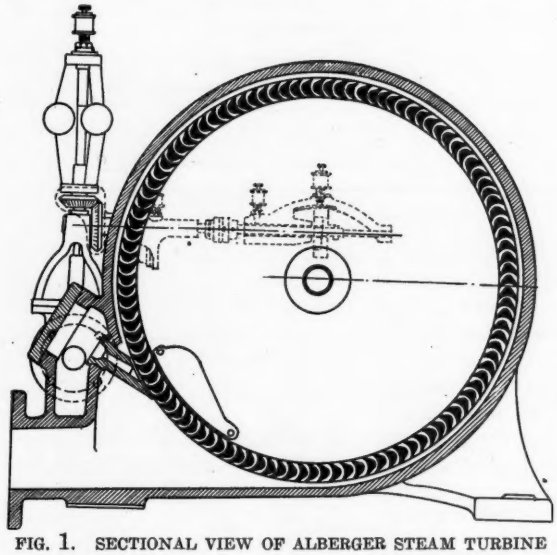


FIG. 1. SECTIONAL VIEW OF ALBERGER STEAM TURBINE

driven through the medium of a lay shaft and connecting gears.

Alberger-Curtis turbines are manufactured in single stage form for noncondensing service only, and consist of a single pressure stage with several velocity stages.

Steam is expanded from initial pressure and temperature to exhaust pressure and temperature by means of expansion nozzles which guide it against the comparatively slow-moving rotor vanes which reverse its direction and reduce its velocity, thereby giving up a portion of the energy to the wheel. On leaving the wheel blades, the steam is received in stationary blades, again reversed in direction and properly guided against the second set of rotating blades, where an additional portion of the available energy is extracted. This operation is again repeated until the steam has passed through sets of rotating blades when practically all of the available energy of the steam has been absorbed.

The casings of these turbines, to which are bolted the bearing brackets, are of the split annular form, divided on the horizontal center line so as to allow removal of the upper half without disturbing any pipe connections. Steam and exhaust connections are cast integral with the lower half, which also contains the nozzles and nickel bronze stationary blades, held in dovetailed grooves in an arc secured to the lower part of the casing.

Chrome nickel steel is employed in the construction of the rotor. To this are secured by means of dovetailed grooves on the rim, nickel bronze blades to which in turn are riveted shroud rings.

The bearings employed are of the ring-oiling type fitted with oil reservoirs and oil sight gages, and are provided with removable caps and split bushings. A multi-collar, marine type thrust bearing with an adjustable device is incorporated in the outboard bearing to maintain the rotating element in correct lateral position. A deep stuffing box is provided at either side where the shaft passes through the casing.

The governor is of the Pickering type, driven from the main shaft through worm gearing located within the outboard bearing body. An externally adjustable

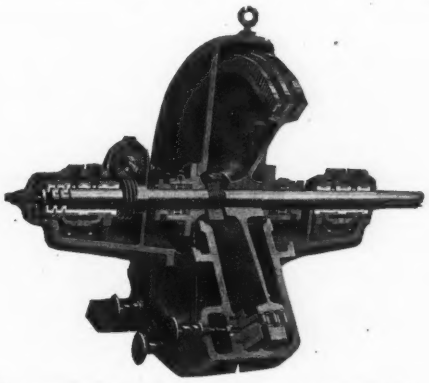


FIG. 2. INTERIOR VIEW, SHOWING CONSTRUCTION OF ALBERGER-CURTIS TURBINE

spring is provided by means of which a moderate variation of speed may be obtained above or below normal rate.

Allis-Chalmers Steam Turbines

OF PARSONS reaction type, these machines, which are supplied in sizes from 300 kw. up, have the reduction in steam pressure divided into a great number of stages of pressure drop, the reduction seldom exceeding 3 lb. in any row of blades.

As seen in Figs. 1 and 2, steam, passing through regulating valve D, flows by pipe C to space E, thence through the guide and revolving blades to exhaust space F and outlet G. Steam flows through the full rings of blades between the spindle and cylinder.

The spindle, as a rule, has three steps, and blades in each step are in groups of increasing length to provide for increase in volume of the steam as it expands, the blades at the beginning of each step being usually shorter than at the end of the smaller preceding step.

As the steam produces an end thrust toward the left, balance pistons L, M, N are provided to counteract the effect on steps H, J and K, respectively, with

with corresponding stages of the blading. Alternate rings on cylinder and pistons form a labyrinth packing to retard leakage of steam. The spindle thus floats, and is easily moved endwise, hence to fix the position of the balance piston rings with reference to the dummy rings in the cylinder and to fix the endwise clearance of

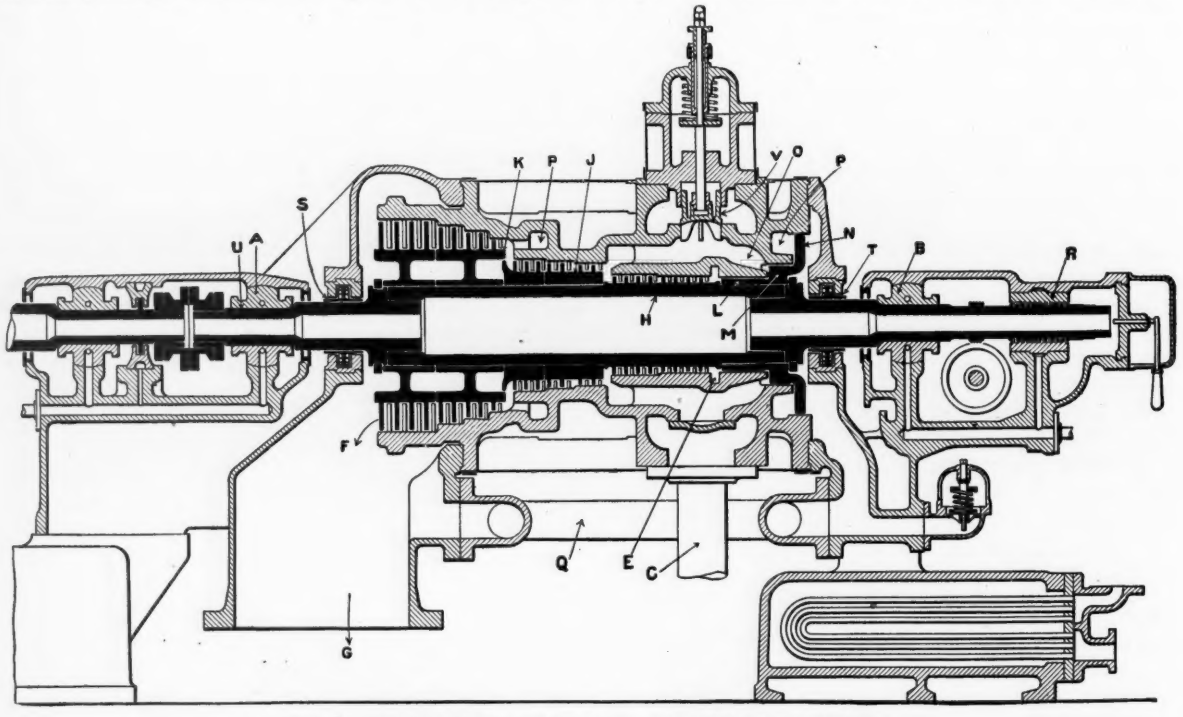


FIG. 1. LONGITUDINAL SECTION OF ALLIS-CHALMERS TURBINE

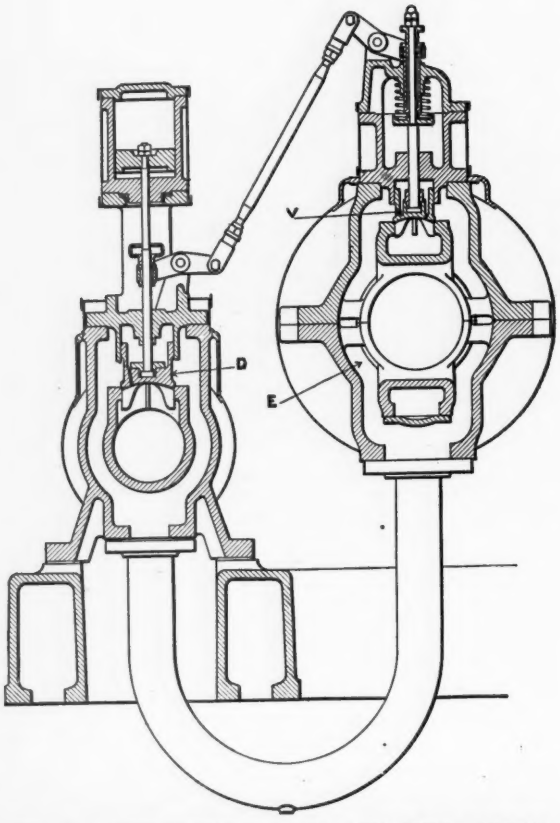


FIG. 2. CROSS-SECTION OF ALLIS-CHALMERS TURBINE

revolving blades, a collar bearing, R, is provided with adjusting screw and lock nut. In smaller sizes, only the upper half of the bearing is movable; in larger sizes, both upper and lower halves are movable and adjustment which because of large bearing surfaces is needed only at long intervals is intentionally made difficult to avoid tampering.

At the generator end, the shaft is extended and joined to the generator shaft by a flexible coupling, Fig. 4, hav-

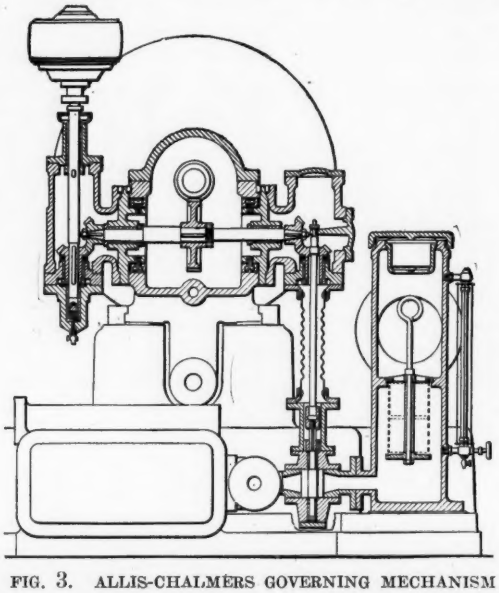


FIG. 3. ALLIS-CHALMERS GOVERNING MECHANISM

ing one section keyed to the end of each shaft, and the sections joined by splines.

Where the shaft passes out of the cylinder, outleakage of steam or inleakage of air are prevented by labyrinth packings and a paddle wheel gland which is water

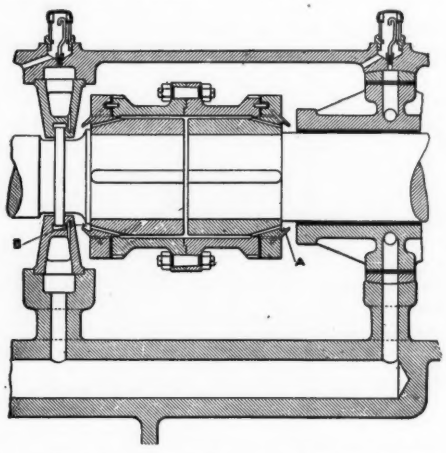


FIG. 4. FLEXIBLE COUPLING OF ALLIS-CHALMERS UNITS

packed as indicated in Fig. 5. This cut shows also the larger labyrinth packing used to prevent steam leakage between the intermediate stage and exhaust when the low-pressure balancing piston is located at the exhaust end of the turbine as in large sizes.

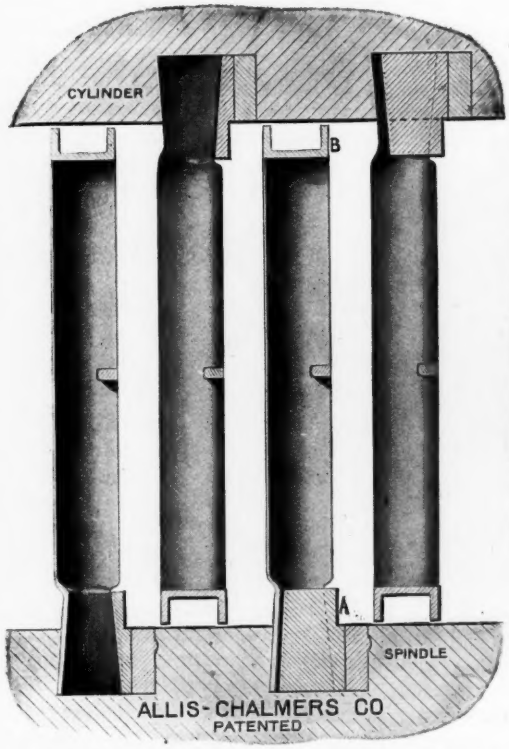


FIG. 6. ALLIS-CHALMERS BLADING AND METHOD OF FASTENING

In the smaller sizes the spindles are of forged steel, having the blade rings installed in the spindle body, while for larger sizes the blade rings are installed in separate forged steel rings which after balancing are pressed on the shaft with taper fits. Each part is balanced separately, and the whole balanced after the final assembling to prevent vibration when running. In balancing end thrust in the larger sizes, the low-pressure balancing piston is placed at the exhaust end of the turbine and acted on by steam from the second

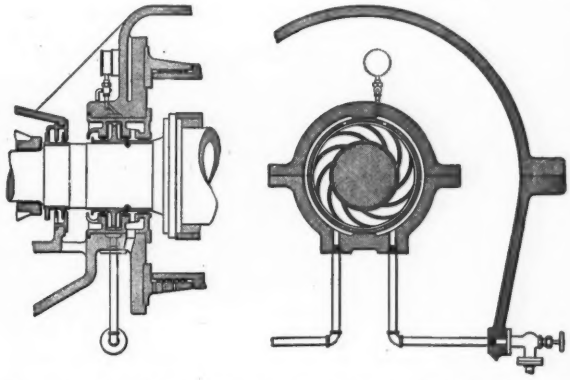


FIG. 5. LABYRINTH AND WATER SEAL PACKING ON ALLIS-CHALMERS TURBINE SHAFT

intermediate passage, entering through the hollow part of the third spindle step. The arrangement is shown in Fig. 5, where exhaust steam is carried in the space next the cylinder and intermediate steam in that next the spindle. This gives a smaller balancing piston, which is also stiffer, as it is backed up by the body of the spindle. The labyrinth packing between this piston and its cylinder is made with small radial clearance and large axial clearance to allow for expansion of the spindle which takes place toward the low-pressure end on account of the collar bearing at the high-pressure end.

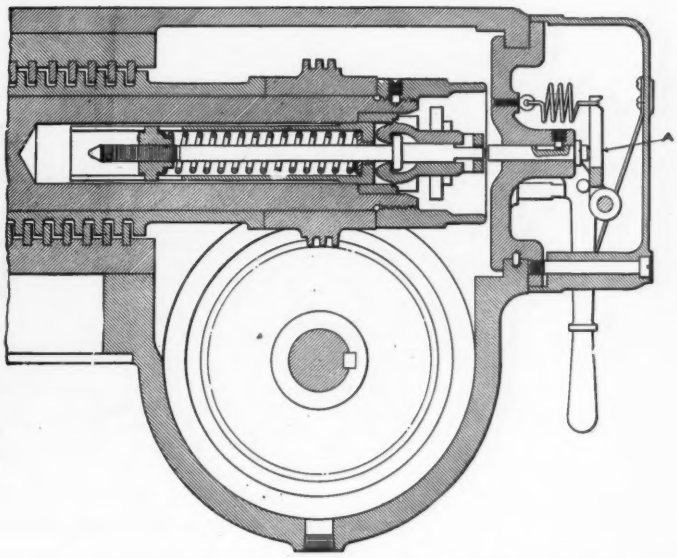


FIG. 8. ALLIS-CHALMERS OVERSPEED GOVERNOR

Blades are individually formed by special machinery with dovetail roots which fit into slots in foundation rings A, Fig. 6. The rings are formed to a circle of proper diameter and dovetailed slots milled to receive

the roots of the blades holding them at the right pitch and angle. Tips of the blades are held at proper angles and protected by a shroud ring, B, which also prevents erosion of the walls of the cylinder.

To fasten the foundation rings to cylinder and spindle, they are made of dovetail shape to fit into dovetail grooves and key pieces are then driven and upset into undercut grooves, thus locking the whole structure together. The blades are thus firmly held without dependence on friction or gripping action of calking pieces. Flanges of the channel shroud rings are made thin so that in case of accidental contact with spindle or cylinder no dangerous heating will ensue. Also, by this means, smaller working clearances are made possible, thus reducing leakage loss, which is also minimized by the baffling action of the channel flanges.

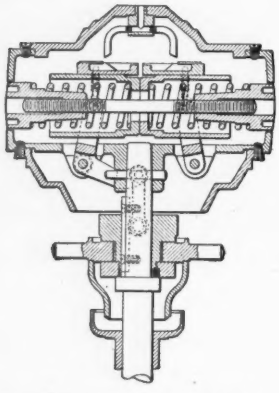


FIG. 7. DETAIL OF ALLIS-CHALMERS GOVERNOR

Speed regulation is furnished by a governor located and driven as shown in Fig. 3. The governor has weights acting horizontally against the force of coiled springs and, by means of bell cranks and linkage moving a sleeve sliding on the vertical spindle, Fig. 7. This operates a relay valve controlling a balanced throttle valve and thus regulating pressure in the steam chest. The governor is lubricated while in operation through a hole in the top of the casing and can also be adjusted for speed while running.

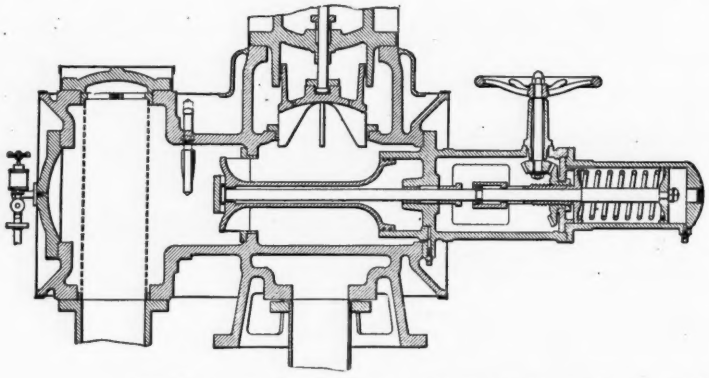


FIG. 9. CONSTRUCTION OF ALLIS-CHALMERS THROTTLE AND GOVERNOR VALVES

To provide against accident, an overspeed governor is fitted to the end of the shaft, Fig. 8, to strike a pin mounted in the end of the pedestal and release a trigger, A, and trip weight which trips the main throttle allowing it to close and shut off steam.

In order to secure best economy at rated load, the first row of blades in the smallest step is designed for that load. The turbine cannot, therefore, carry all possible overloads when taking all its steam through those blades. To provide for this, an overload valve is fitted, V, Fig. 1, which is operated by the governor and admits live steam to the first intermediate space, O, when

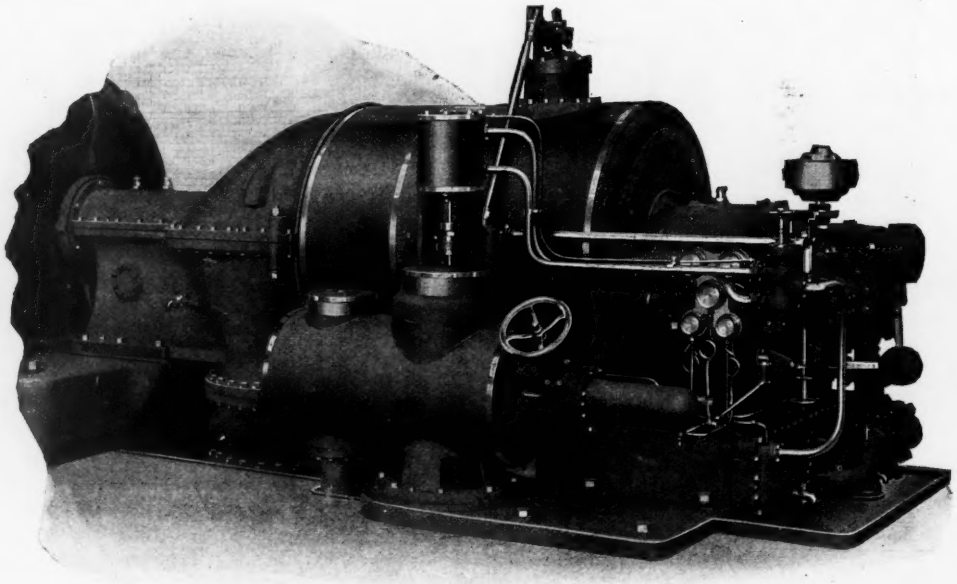


FIG. 10. A 2500-KW. ALLIS-CHALMERS STEAM TURBINE

speed falls below a desired minimum, due to the load being greater than the turbine can carry in normal operation. This greater power is obtained at a temporary sacrifice of economy and is only intended to carry over emergency peaks, not for regular operation.

Construction and relative positions of governor valve and throttle valve are shown in Fig. 9. Hand wheel control acts on the sleeve carrying with it the spindle, spring and bearing heads. When tripped, the spring forces the right-hand head to the right, drawing the spindle through the sleeve and closing the valve. At the left is seen the strainer, dotted, which has openings of such area as to allow free flow of steam, but to stop anything which would be of size to damage the turbine blades.

Heated parts of the turbine are covered with nonconducting material and lagged with planished steel, so applied that it can be easily removed, also the nonconducting material at the horizontal joint where the cylinder is split to open the turbine for examination.

For small sizes, a continuous bed plate is used, and the turbine shipped as a complete unit. For large sizes, the bed plate is divided, one part carrying the generator and low-pressure end of the turbine, the other the high-pressure end. The turbine is secured to the former, slides in guides on the latter to allow for expansion while maintaining alinement. The space between foundation piers is used for condensing apparatus.

For low-pressure work, the Allis-Chalmers turbine is similar to the high-pressure, but has fewer stages and larger blades. All steam is passed through all blades, no provision being made for admitting steam at a later stage. Live steam may, however, be admitted by the governor through a special throttling valve for emergencies when the low-pressure steam supplied is insufficient to carry the load. Figure 10 shows the assembly of a 2500-kw. machine to run at 2400 r.p.m.

Instructions for the Operation of Allis-Chalmers Turbines

By J. Wilson

As it is impossible to discuss the operation of all types of steam turbines manufactured by the Allis-Chalmers Manufacturing Co. within the limits of this article, it is the intention of the writer to confine his remarks to the type most commonly found in practice, namely, the high-pressure condensing type. With slight modifications an experienced engineer can apply the rules set forth to apply on noncondensing, low-pressure and mixed-pressure units.

I appreciate that the instructions suggested below for starting are open to criticism on account of the length of time recommended for warming up the unit, and I am perfectly aware that this time may be cut to a very few minutes without danger by an experienced engineer, but a manufacturer, when drawing up a set of instructions, must be guided by the ability of the least experienced engineer likely to operate the unit and must draft his instructions so that if they are followed explicitly, the unit can be started up without any possibility of trouble. When the occasion demanded, many Allis-Chalmers turbines have been started up from dead cold and put on the line within 3 or 4 min. without experiencing any trouble; but this method of starting should not be followed as a general practice, and then only by an experienced engineer thoroughly familiar with his apparatus.

The following are the Allis-Chalmers Co.'s instructions covering the starting and operation of a typical Allis-Chalmers high-pressure condensing unit:

Starting

1st. Start the auxiliary oil pump. Make sure its speed is high enough to give an oil pressure sufficient to lift the inlet valve, and for oil to flow steadily at the vents on all bearings. (The oil pressure required for the above is about 25 lb. on the "Relay Oil" gage, and 2 to 4 lb. on the "Bearing Oil" gage.)

2nd. Turn the cooling water on the oil cooler and make sure there is an ample steady flow.

3rd. Open all steam drains on turbine and steam piping.

4th. Open throttle valve as far as can be done without unit turning over and let steam blow through for 15 min., then start spindle turning slowly (about 100 r.p.m.) and allow turbine to run at this speed for 10 min.; after this bring unit gradually up to speed. After unit is up to speed and under control of governor, open throttle valve wide.

5th. Turn the water on to the glands gradually and just enough to seal properly. When operating condensing, water seal drain valves should be closed. When operating temporarily noncondensing, water seal drain valves must be open enough to allow circulation of water through the glands sufficient to prevent gland water steaming.

Do not shut off auxiliary oil pump until turbine is almost up to speed. Make sure the oil pressures are kept up by the gear driven oil pump on the turbine when the auxiliary oil pump is shut off.

Close steam drains after there is no flow of water from them.

6th. Build up the vacuum gradually to about 25 in. of mercury.

7th. Put on the field current and adjust rheostat for normal voltage.

8th. Synchronize carefully if parallel operation is required.

9th. Build up the load on the unit gradually. Do not carry more than 25 in. of vacuum when load is below 0.1 of normal rating.

Don't operate without oil flowing through all bearings.

Don't operate without water on cooler.

Don't turn steam on turbine unless water is circulating through condenser.

Note—Local conditions generally make starting noncondensing preferable. Unit may be started condensing, in which case do not let vacuum exceed 25 in. of mercury with no load on unit.

Operating

Watch the following:

In case the pressure of "Relay Oil" and "Bearing Oil" decreases appreciably, start the auxiliary oil pump

immediately; then locate and correct the cause of the trouble.

There must be a free flow of oil at the vents on *all* bearings. If the temperature of oil *from* bearings exceeds 150 deg. F., there is probably something wrong with oil cooler or oil, and steps must be taken at once to correct the trouble. Oil will lubricate best when temperature of oil from bearings is between 110 and 140 deg. F.

Oil the governor levers frequently.

Moisten the collector rings with a little vaseline when necessary.

Stopping

1st. Start the auxiliary oil pump slowly.

2nd. Remove the load gradually and reduce the vacuum to about 25 in. of mercury when load reaches about 0.1 normal load.

3rd. Pull the trip lever, which will close the throttle valve. As the speed of the turbine decreases, speed up the auxiliary oil pump to maintain the proper oil pressures.

4th. Break the vacuum gradually.

5th. Shut off the gland water.

6th. After the turbine has stopped, shut down the auxiliary oil pump; shut off the cooling water and open all steam drains. Oil the valve stems slightly with cylinder oil. Close the throttle valve and hook up the trip.

DON'T SHUT OFF STEAM WITHOUT STARTING THE AUXILIARY OIL PUMP.

Remarks

Water: The water used for the glands of the turbine must be free from acid and scale-forming impurities and delivered to the glands at a steady pressure. The pressure at the glands will vary from 8 to 15 lb. and the temperature should not exceed 100 deg. F. Condensed steam from a surface condenser is satisfactory for this purpose.

The water for the oil cooler must be cold, clear and free from dirt and at a steady pressure not exceeding 15 lb.

Oil: The oil used in the turbine must be absolutely pure mineral oil of a non-foaming character, free from animal or vegetable fats, paraffin and water.

When filling the oil tank, strain the oil through a cheese-cloth bag. Keep the oil tank about ⅔ full.

Clean the oiling system periodically, draining all the oil from the turbine, and filter it before using again. New oil should be added as required.

Condenser

When a surface condenser is used, see that water is circulated through the condenser tubes when the turbine is started or operated temporarily noncondensing; this will prevent baking the tube packing. Do not allow an excessive amount of condensate to accumulate in the bottom of the condenser.

If the installation includes a jet condenser and the turbine is operating temporarily noncondensing, see that the condenser pumps are submerged in water. This will prevent burning the shaft gland packing. Also make sure that the vacuum breaker is always in working condition and that the water in the condenser does not rise above normal level.

If a barometric condenser is used, see that the exhaust main is kept free from water by means of a high vacuum pump or vacuum drainer.

Boilers

Avoid slugs of water; a slug of water may seriously damage the spindle and blading.

For a better understanding of the above instructions and the order in which they are set forth, it may be advisable to go a little more into detail.

Starting

1st. When the unit is shut down, the weight of the inlet valve, relay piston, etc., causes the inlet valve to close; therefore, when again starting up, before steam can enter the turbine sufficient relay oil pressure must be built up to open the inlet valve; furthermore, the spindle must not be started rolling until all the bearings are properly lubricated.

2nd. The oil, in addition to lubricating the wearing parts, is the medium for removing the heat generated by the pressure on the bearings and high speed of rotation; hence, the necessity of the oil cooler to prevent building up the temperature of the oil.

3rd. Before starting, the piping, steam chest, etc., are cold; therefore, immediately the steam is turned on, considerable condensation takes place. This water must be prevented from entering the turbine, as an excessive quantity of water flowing along the bottom of the cylinder would cause uneven heating of the cylinder.

4th. The cylinder and spindle do not heat up at the same rate, due to the exposed surfaces of the cylinder and to the heavy cylinder flanges and ribs; consequently, whenever possible, sufficient time should be allowed after turning on the steam and before starting the spindle rolling for the cylinder to be thoroughly heated through and as nearly at its normal operating temperature as possible.

The spindle should be started turning slowly, so that it is evenly heated throughout before it is brought up to speed. If this is omitted and the unit is brought quickly up to speed after being heated by blowing steam through the cylinder, the chances are the spindle will be heated more on the top than on the bottom. Due to the difference in the expansion in the two sides, the spindle will buckle slightly and become temporarily unbalanced. The consequent vibration may cause blade fouling and result in serious injury to the apparatus.

5th. The water seal glands, where the spindle passes through the ends of the cylinder, are in effect small centrifugal pumps with the discharge outlet closed in the case of the condensing unit, or if the machine is operating temporarily noncondensing, with the outlet open just enough to permit sufficient flow of water through the glands to prevent steaming. The water inlet pressure should be slightly in excess of the pressure generated by the runner, i. e., sufficient to keep casing full of water and to make up for any leakage into the turbine. An excess pressure is not only wasteful of water but may be harmful, if the pressure is sufficient to cause the excess water to strike the back of the balance piston, thereby unduly cooling one side and causing buckling or

deflection of the balance piston and possibly fouling of the dummies.

It is important that the water for sealing purposes be free from acid or scale-forming impurities. Acid will corrode the runners and casings, and this in turn will reduce the efficiency of the seals to such an extent air will be permitted to leak into the vacuum space and so reduce the overall efficiency of the installation. Scale forming impurities in the water will have the same result. The scale will be deposited in between the vanes of the runners, sooner or later rendering them ineffective; hence, it is necessary to supply the glands with pure water. Condensed steam from a surface condenser or from steam pipes or heating coils is satisfactory for this purpose. It is a simple matter to install a small water purification system if satisfactory water cannot be obtained in any other way.

6th. It is usually undesirable to operate a machine with full vacuum and no load, as under these conditions vacuum temperatures extend almost to the inlet of the turbine, unduly cooling down the machine. When load is thrown on, the sudden increase in temperature due to the inrush of steam sets up severe stresses in the cylinder. The writer fully appreciates that conditions of operation will sometimes prevent this rule being followed, but these exceptional cases should not deter the careful engineer from taking the best care of his unit whenever possible.

As the steam turbine is automatic in its action, the operator's duty is to observe that the parts are functioning properly, that nothing interferes with the steady flow of water to the glands and cooler and that the steam pressure and vacuum are maintained at the desired point. Any unusual occurrence should be investigated promptly and any trouble rectified at once. A small trouble neglected may result in serious injury to the unit and cause a long shutdown for repairs.

A frequent source of trouble is loss of blading, due to slugs of water entering the turbine with the steam. I am aware that in many cases a slug of water entering the turbine will simply slow down the unit until the water drains away through the exhaust; but sooner or later such operation will result in the loss of blades, a long shutdown and considerable expense. Therefore, slugs of water should be avoided. Blading is also sometimes lost due to water backing up in the exhaust pipe on account of flooding the condenser, or to the accumulation of water in the exhaust pipe due to faulty layout of the exhaust piping. In the latter case the water gets surging to and fro until finally it enters the turbine exhaust, and damage is caused by the blades, traveling at high velocity, striking the water. Careful operation of the condensing plant and care in the layout of the exhaust piping is essential.

There are very few wearing parts in a steam turbine; but to insure continuous service, a thorough inspection of the apparatus should be made at least once every year.

De Laval Steam Turbines

THESE are made in single and multi-stage arrangements, the impulse principle being used in both. Construction details are similar, the single stage being used for smaller powers, so that only the multi-stage will be described.

Where the speed of the driven machine approximates that of the turbine, for good efficiency, direct connection is made. Otherwise helical reduction gearing is used so that both turbine and driven machine run at the speed best adapted to secure high efficiency.

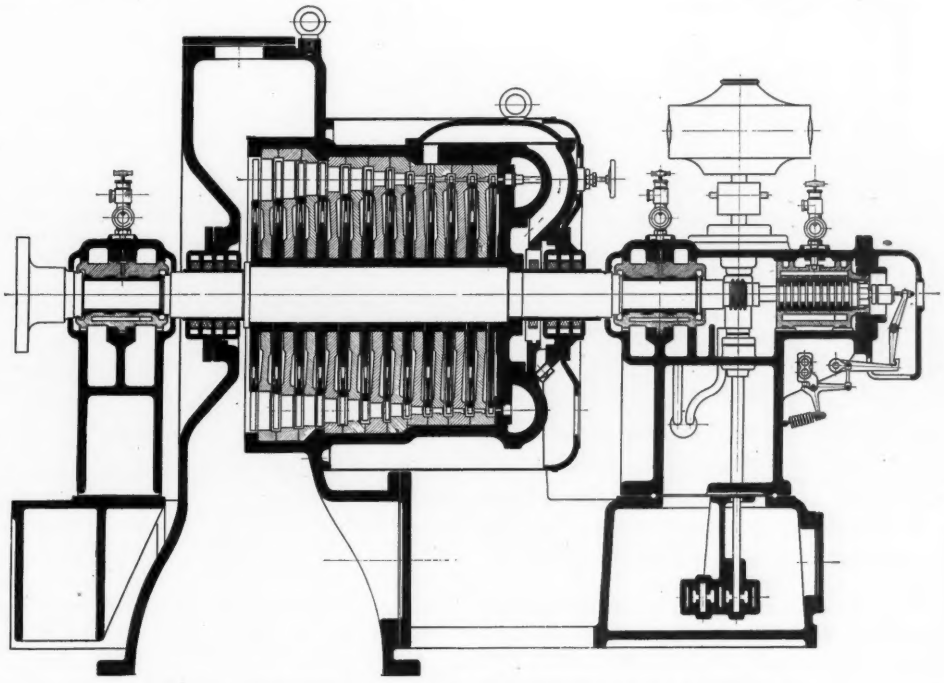


FIG. 1. ARRANGEMENT OF DE LAVAL MULTI-CELLULAR TURBINE

De Laval turbines are built for all conditions of initial and exhaust pressures. In large powers and for high efficiencies the multi-cellular arrangement of the wheels is used, each wheel revolving in its own chamber and carrying a single row of radial blades. An exception is to be made of the first or highest pressure stage, the wheel of which usually carries two rows of impulse buckets, between which is a row of stationary guide vanes. For sizes up to 200 hp. a single wheel of single stage type is advocated for condensing operation on account of the high efficiency attainable, unless speed conditions made the multi-stage type preferable. The latter type is built in sizes from 200 to 5000 hp.

Figure 1 shows the general arrangement, with a series of disks mounted on a heavy shaft. Each disk revolves in its own cell formed by diaphragms held in a cylindrical casing. Steam enters at the right hand end and flows successively through nozzles and blades to the exhaust chamber at the left.

Nozzles for the first stage are either bored and reamed directly in the nozzle ring or are set in. They cover only a portion of the circumference of the first wheel, thus permitting the use of blades of proper length, from a

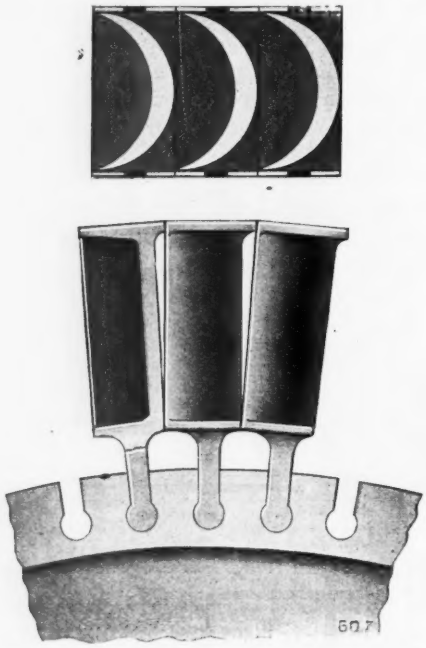


FIG. 2. FORM AND METHOD OF ATTACHING DE LAVAL BUCKETS

mechanical standpoint, in that stage, and some of the nozzles may be controlled by a hand-operated valve, seating upon the inlet opening, to adjust the amount of steam admitted to changes in load.

Buckets, which are of a special alloy, are formed by drop forging, giving a smooth, hard, glossy coating to resist the action of steam, water and corrosive acids, also high tensile strength and uniform tough texture. On the tips, as shown in Fig. 2, lugs are formed which abut on adjacent buckets, forming a continuous rim, thus reducing fan action, confining the jets of steam within closed channels, maintaining proper spacing, and preventing vibration of the blades.

Transverse dovetails secure the buckets to the rim of the wheel, so that any blade may be replaced without damaging others. The round part of the slot is drilled and reamed and the straight part accurately milled, giving a perfect fit and a strong connection.

Increase in area of passages to provide for greater volume of steam as it expands is provided by lengthening the blades, reducing the diameters of the wheels and increasing the bore of the casing correspondingly, as shown in Fig. 1. The length and strength of blades on the last wheel determine the maximum speed at which the turbine may be operated.

The wheels are forged steel disks, finished on all surfaces, with hubs extended so that adjacent hubs touch, permitting all wheels to be locked in place by a single nut, and adding to the stiffness of the shaft. The wheels are mounted on the shaft by means of taper bushings, Fig. 3, thus centering them and permitting of easy removal should that ever become necessary.

Except in the first stage, the nozzles for turbines above 1500 hp. are formed between bronze guide vanes,

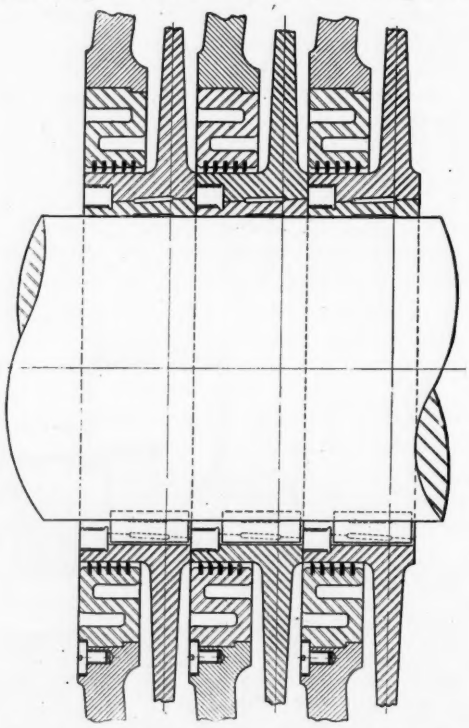


FIG. 3. ATTACHMENT OF DE LAVAL WHEELS TO SHAFT

accurately formed in dies, hammered to give polished surfaces and density of metal, and set around the entire rim of the diaphragm. They are located on the rim by two pins to give the proper angle, and held in place by a solid steel band shrunk over their tips. As shown in Fig. 1, these bands are wider than the blades, so that adjacent bands touch and form a continuous cylinder, enclosing both diaphragms and wheels, and giving a lining of forged steel for the casing.

At the center, the cast-iron diaphragms are perforated and fitted with removable labyrinth packings, Fig. 3, to reduce the leakage of steam from stage to stage between the diaphragms and wheel hubs. The heavy rings outside the guide vanes rest in a cast-iron casing

which is split horizontally to give access to the working parts. The case is supported in the plane of the center line on two pedestals rising from the bed plate, which permits expansion diametrically and lengthwise, as all holding down bolts but one on each side have oval bushings and are shouldered where they screw into the pedestal. The casing can, therefore, remain at all times concentric with the shaft.

To reduce vibration and consequent leakage, the shaft is made large and stiff so that its critical speed is

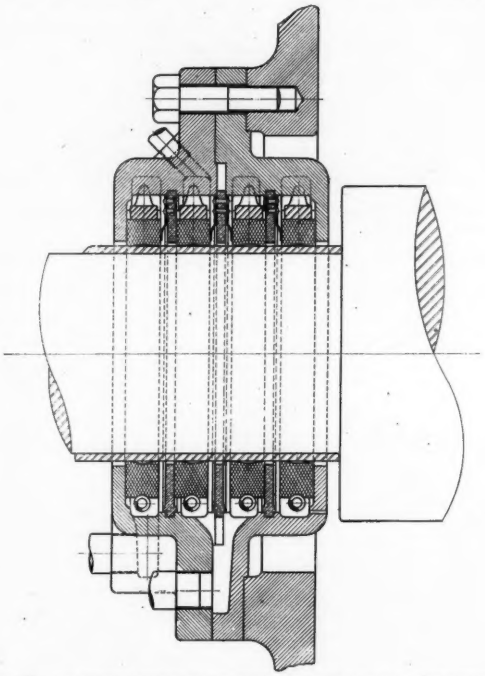


FIG. 4. CARBON RING PACKING ON DE LAVAL TURBINE

higher than the running speed, even neglecting the stiffening action of the wheel hubs. As the turbine is of moderate length, the bearings can be placed comparatively close together, which also reduces the tendency to vibration.

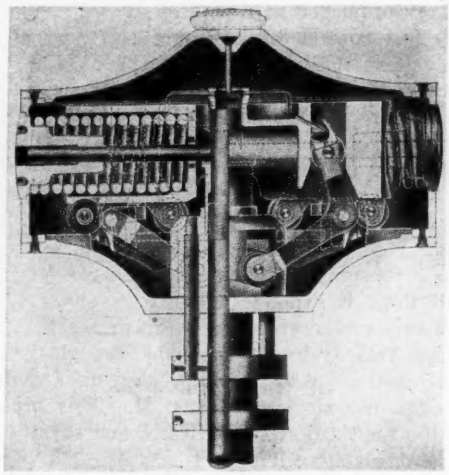


FIG. 5. DE LAVAL GOVERNOR

Leakage between stages is minimized by the labyrinth packing shown in Fig. 3. That from the first cell to atmosphere is obstructed by a double length labyrinth and any steam getting past is returned to an intermediate stage and used at a lower pressure in the following stages, as indicated in Fig. 1. Beyond the labyrinth is a series of segmental carbon packing rings, each one in a separate compartment. Leakage past the first ring is carried to a low pressure stage or to exhaust, and low pressure steam is introduced between the two outermost rings so that any inward leakage will be steam rather than air. At the exhaust end, similar carbon rings are used, as seen in Fig. 4, where the live steam connection between outermost rings is shown.

Three bearings are used, two to support the weight of the revolving parts, and the third a thrust bearing to maintain the endwise position of shaft and wheels, and determine clearances. The main bearing shells are of cast-iron, lined with babbit, split vertically to relieve the caps of strain. Bearing seats are scraped and out-

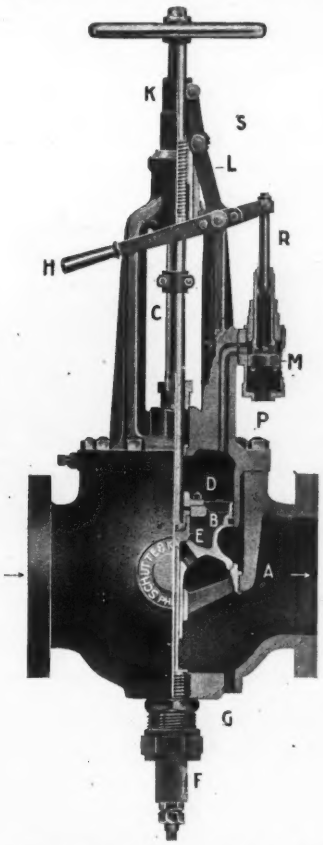


FIG. 6. DE LAVAL EMERGENCY GOVERNOR ARRANGEMENT

sides of shells ground to ensure alinement and interchangeability.

For controlling the speed a governor, Fig. 5, is mounted on a vertical shaft and driven from the turbine shaft by worm gearing. It controls the admission of steam to the steam chest. Weights, moving in a horizontal plane, and guided vertically and horizontally by roller bearings, act directly against springs in such manner that the pressure is not carried by the governor mechanism. These weights, by means of bell cranks with roller bearing connections, control a sliding sleeve on the spindle, which in turn operates the throttling valve. Parts are so designated and arranged that the governor possesses constant energy for any position of the weights. Upper pins and slides are oiled through a top oil cup

and lower pins and slides run in a bath of oil, the overflow lubricating the spindle sleeve. The link system connecting to the valve contains a spring which prevents injury in case of overtravel of the governor. The governor valve, which is between the turbine and the main throttle, is of poppet type, balanced and double seated with adjustable valve disks and renewable seats.

To provide against racing, due to accident to the main governor or its drive, a speed-limiting safety stop and trip is provided on the end of the main turbine shaft, as shown in Fig. 1. At the speed for which it is set, this stop trips a small valve, releasing steam pressure at M from under a small piston in the throttle valve, Fig. 6, so that the pressure admitted to D forces the balanced disk of the throttle valve to its seat. This valve may also be hand tripped in case of emergency. Fig. 7 shows the exterior and arrangement of a De Laval turbine of multi-stage type.

The De Laval Steam Turbine Co. also builds velocity-stage turbines in all sizes up to 1000 hp. In this type of turbine the steam is expanded from the initial to the final pressure in a single set of nozzles, but the jets strike upon two or more rows of moving buckets in succession, being redirected by rows of stationary blades or guide vanes placed in between the rows of moving buckets on the wheel. This type of turbine is therefore adapted to run at slower speeds than single-stage turbines and will give fairly good economy when directly connected to moderate-speed machinery. Wherever the exhaust steam can be utilized, as for heating boiler-feed water, heating buildings, or drying, its ideal simplicity, ruggedness and reliability strongly recommend it.

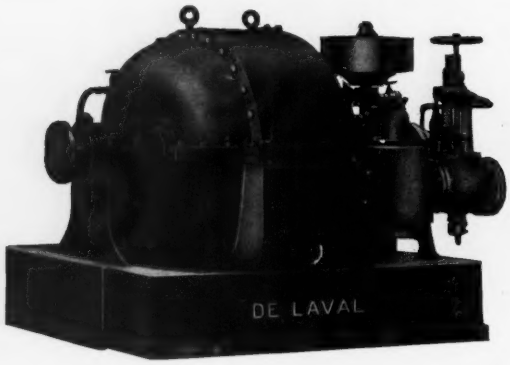


FIG. 7. THE DE LAVAL MULTI-STAGE TURBINE

Two or three velocity stages are used on the wheel of a velocity-stage machine, depending upon steam pressure and capacity. The turbine wheel runs in steam at exhaust pressure and there is therefore little tendency to leakage through the stuffing box at the turbine end of the unit. This unit is ordinarily controlled by a pressure governor of some sort, but is also fitted with an emergency speed-limiting governor, which prevents accidental runaway in case of lack of water supply.

Pressure-stage turbines are built in which each pressure stage contains two or more velocity stages by the De Laval Steam Turbine Co. in sizes up to 2000 hp.

General Electric Turbines

IN THE Curtis type of impulse steam turbine, as built by the General Electric Co., the rotating element is actuated by the impact of steam passing through its buckets at relatively high velocity, the steam being expanded in stationary nozzles, wherein it acquires relatively high velocity and then passes through and actuates the moving buckets of any stage, without further expansion.

Entering at A, Fig. 1, the steam passes from the steam pipe into the steam chest, B, and then through one or more open valves to the bowls, C. The number of valves open depends on the load, and their action is controlled by the governor. From bowls C, the steam expands through divergent nozzles D, entering the first row of revolving buckets of the first stage at E, thence passing through stationary buckets G, which reverse its direction and redirect it against second revolving row H.

This constitutes the performance of the steam in the first stage, or pressure chamber. Having entered

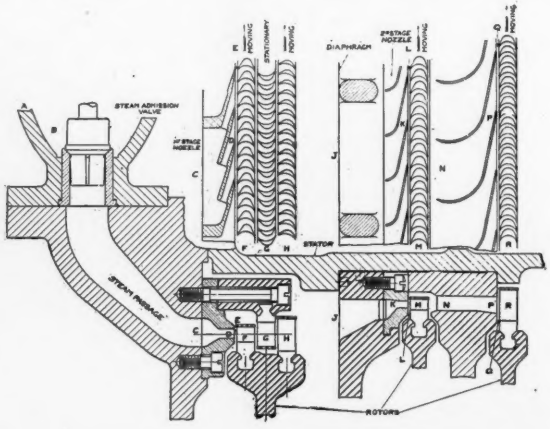


FIG. 1. DIAGRAMMATIC ARRANGEMENT OF MOVING AND STATIONARY ELEMENTS OF GENERAL ELECTRIC CURTIS TURBINES

the first row of buckets at E with relatively high velocity it leaves last row H with relatively low velocity, its energy between the limits of inlet and discharge pressure having been abstracted in passing from C to H. It has, however, a large amount of unexpended energy since the expansion from C to E has covered only a part of the available pressure range. The expansion process is, therefore, repeated in a second stage.

The steam having left buckets H, and having had its velocity greatly reduced reaches a second series of bowls J, opening upon a second series of nozzles K. Through these it expands again from the first stage pressure to some lower pressure, again acquiring relatively high velocity in its expansion through these nozzles, leaving them at L and impinging upon and passing through the moving buckets, M. This process is repeated in the third stage nozzles N, and so on through the remaining stages, if any.

Curtis machines are usually constructed with the first stage wheel carrying two rows of revolving buckets, followed by several stages, each containing a single row.

Small sizes are built with one or more stages, each with two or three rows of revolving buckets.

The wheels employed consist of steel disks, carrying rotating buckets, made of a noncorrosive metal of high tensile strength, on their periphery, these buckets being

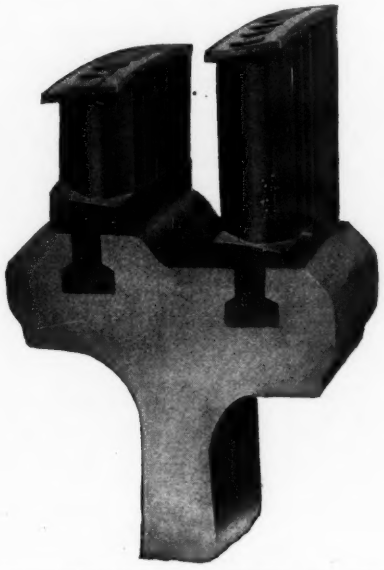


FIG. 2. DIAGRAMMATIC ARRANGEMENT OF BUCKETS OF A SINGLE CURTIS STAGE

held in place by a dovetail shaped root which fits snugly into a channel of the same section, machined in the rim. Figure 2 illustrates this construction, while Fig. 3 shows the wheel rim with a row of buckets partially inserted. The dovetail channel is opened for the insertion of buckets as shown at A. This opening is afterward filled with a spacing block, and closed. After the buckets are assembled, a shroud ring, the function of which is to stiffen the complete row and reduce vibration, and to assist in retaining the steam flow in the proper path, is riveted to their outer ends.

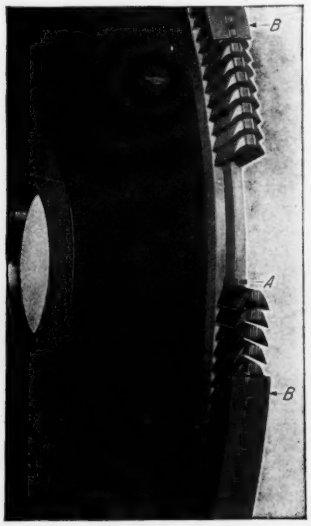


FIG. 3. CURTIS TURBINE WHEEL SHOWING TYPICAL METHOD OF ATTACHING BUCKETS AND SHROUD RINGS

Figure 4 shows a typical section of expanding nozzles. These extend around a relatively short arc of the periphery, in the first stage, while in the low-pressure end they extend around the entire wheel.

The bearings are babbit lined, fitted with oil rings, split and spherically seated and are supplied with oil from a general distributing system, the pressure in which is normally maintained by a geared oil pump, although in the larger machines an independent steam driven pump is also employed.

Where the bearing pressures and speeds so require, positive cooling is effected by pipe coils embedded in the bearing metal, through which water is circulated.

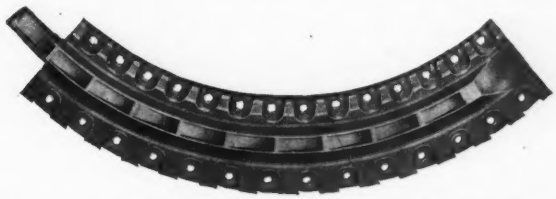


FIG. 4. FIRST STAGE HIGH-PRESSURE NOZZLE PLATE FOR CURTIS STEAM TURBINES, SHOWING EXIT SIDE OF NOZZLES

A simple thrust bearing holds the rotor in a definite position, and prevents axial movement, and in order to prevent the escape of steam along the shaft, steam sealed glands are provided.

A governor of the centrifugal type transmits its motion through a connecting rod to the valve mechanism, which in turn actuates the valves, the valves opening and closing as called for by the condition of the load, an arrangement which practically eliminates throttling.

In addition, a simple speed limiting device is provided to prevent acceleration beyond a certain predetermined limit over normal speed. This consists of a ring unevenly weighted, attached to and revolving with the shaft. At normal speeds and less, the unbalanced ring is held concentric with the shaft by suitable springs, but at excess speed, the centrifugal strain overcomes the spring tension, and the ring then revolves eccentrically, and trips the valve mechanism, causing the main throttle valve to close instantly, thereby shutting off the steam supply.

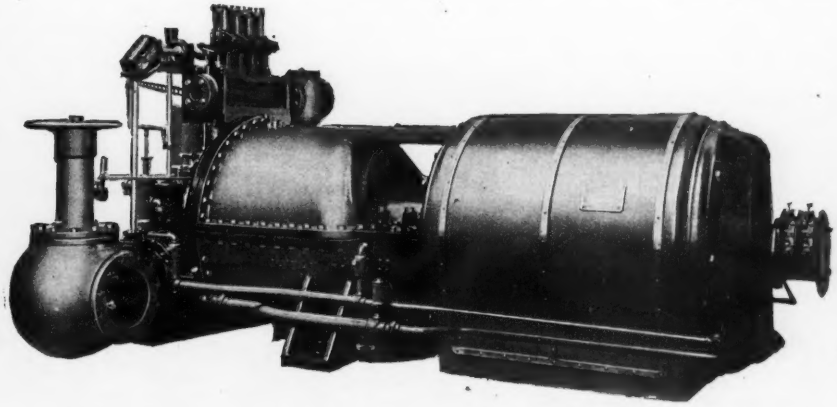


FIG. 5. GENERAL ELECTRIC 500-KW. MIXED-PRESSURE TURBINE

A wire screen supported by cast steel framework, protects the turbine from any foreign material which may find its way into the pipe line.

Curtis mixed-pressure turbines, designed to take the exhaust steam from engines, pumps, air compressors, hoists, etc., and convert it into useful power, may be divided into two main groups. First, those which are essentially low-pressure turbines, designed for high economy when operating on exhaust steam, but provided with special nozzles in which the steam is expanded from boiler pressure and acts directly on the same bucket wheels as the low-pressure steam. These nozzles are automatically brought into action in case the supply of low-pressure steam is for any reason insufficient for the power required for the turbine.

The second group comprises turbines designed for conditions where the supply of low-pressure steam is so intermittent or insufficient as to render economy with boiler pressure steam of first importance. These turbines have a separate high-pressure element which, in conjunction with the low-pressure element, gives good economy when operating on boiler pressure steam. This element runs idle when the turbine is operating on low-pressure steam, but the turbine parts are so arranged that it runs in a vacuum instead of in steam at atmospheric pressure. That is, the high-pressure economy is obtained at the least possible sacrifice of low-pressure efficiency.

The Curtis turbine is in general well adapted to the extraction of steam from one of the early stages for heating or manufacturing purposes. In some cases this is accomplished by taking steam from an opening provided in the first stage; the supply being controlled by an external valve. In other cases an internal valve controls the flow of extracted steam. This valve may be arranged either for manual or automatic regulation.

Where speed reduction is necessary as in the driving of direct current and small low frequency generators, ship propellers, pumps and other mechanical apparatus, a speed reduction gear of the flexible type is employed.

Kerr Economy Turbines

KERR Economy turbines are of the multi-pressure stage impulse type and are built for high-pressure condensing or noncondensing, low or mixed pressure service. Machines of the bleeder type are also available.

The casing employed is made up of steam and exhaust ends which are provided with supports for bolting the whole to the bedplate and between which are the circular steel or iron diaphragms containing the nozzles and dividing the turbine into separate stages.

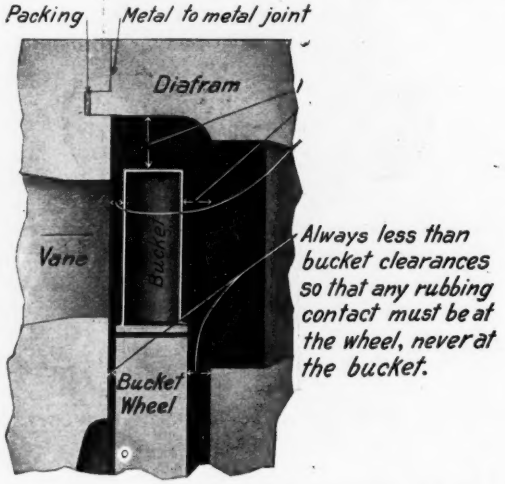

Packing
Metal to metal joint
Diafram
Vane
Bucket
Bucket Wheel
Always less than bucket clearances so that any rubbing contact must be at the wheel, never at the bucket.

FIG. 1. SECTION SHOWING VANE, BUCKET, WHEEL AND DIAPHRAGM

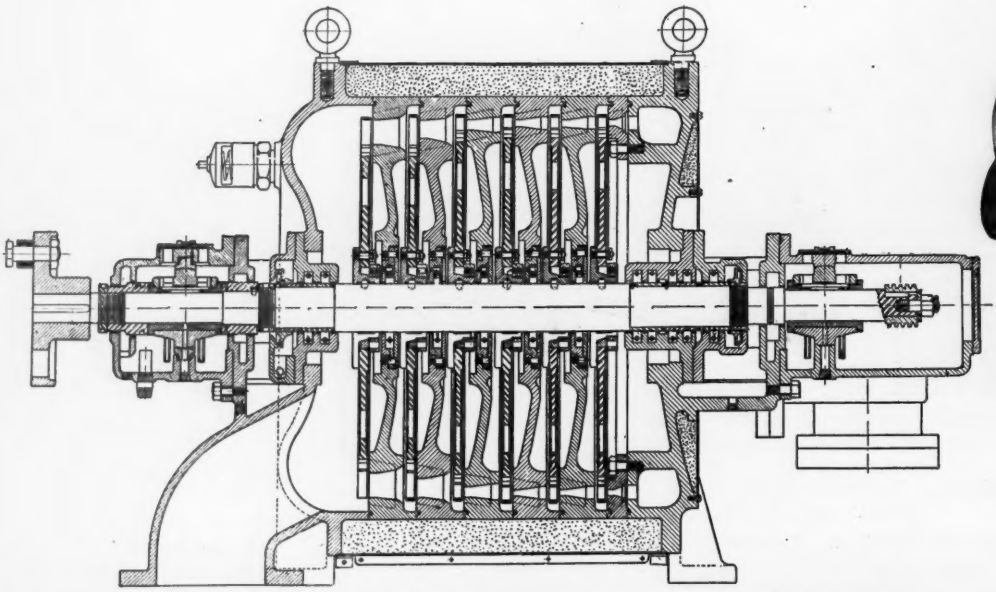


FIG. 2. SECTIONAL VIEW OF KERR ECONOMY TURBINE

FIG. 3. METHOD OF ATTACHING BUCKET WHEELS TO SMALL TURBINE SHAFTS

Diaphragm castings are arched at the center and are centered with each other and with the two end castings by turned and bored tongue and groove joints packed with fibrous packing laid in graphite and oil. The sections are bound together by continuous staybolts passing through drilled holes in the flange of one end casting and tapped into the flange of the other. On split case machines, the diaphragms are bolted together with the exhaust end, forming a cover which may be lifted, exposing the entire rotor for inspection or repairs and in the smaller sizes the end castings are each bored to receive the main bearing cases.

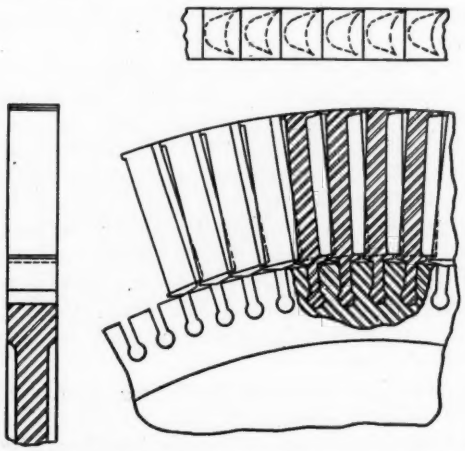


FIG. 4. METHOD OF INSERTING AND HOLDING BLADES

The fact that a diaphragm can be added or removed from this type of turbine without any other change except for lengthening or shortening the shaft makes it exceedingly flexible. The thermo-dynamic efficiency of any turbine, leaving windage losses out of the question, is, as is commonly known, a variable of the relation between blade and steam velocity. By adding stages for exceptional conditions; low rotative speeds, high steam pressure and good vacuum conditions, and removing stages for conditions in the opposite extreme, high rotative speeds, low steam pressures, and high exhaust pressures—it is possible so to adjust the number of stages as to get good efficiency even under difficult conditions.

As a result of this inherent feature in the Kerr-Economy Turbines, they are particularly well suited to work where a great number of different conditions of service are demanded of one machine. So, for instance, the condensing turbines can, at a slight additional cost, be provided with bypass valves to carry full load noncondensing even at a lowered initial pressure.

Steam is admitted through a double-beat balanced poppet valve the stem of which is connected through levers to the governor, two general types of which may be provided according to the nature of the service. For the small machines, a standard direct-acting governor mounted on the shaft and controlling the movement of the steam valve by a lever connection is usually furnished, while in the larger sizes it is customary to provide a governor of the oil relay type. This, instead of being direct acting controls the movement of the steam valve through a small pilot valve which admits oil to either side of an oil relay piston directly connected to the main steam valve. These two types of governors are shown in Figs. 5 and 6.

Wherever necessary, an overspeed governor is used to shut down the turbine automatically in case the speed exceeds a certain predetermined value.

The steam chest has flanged or screwed connections for the throttle valve or steam pipe and is bolted to the steam and casting which is provided with ports for admitting steam to the first stage nozzles. Steam leaves the turbine through a flanged opening cast integral with the exhaust end.

On rotors of the larger sizes, the bucket wheels are forged integral with a central hub after which they are forced by hydraulic pressure against each other and onto a tapering chrome vanadium steel shaft. When the whole is assembled, a two-part circular key is inserted and held in place by a shrink ring.

In the smaller units each bucket wheel is bolted into a three-piece split hub fitted to the shaft and kept from endwise or rotary movement by pin keys as shown in Fig. 3. A split mitered key is forced by a lock nut into the bore of the disc and against a turned taper on the hub.

Drop forged noncorroding metal buckets are inserted sidewise into the rim of the wheel by means of bottom shanks and are then riveted into place. The wheels are machined from flange steel or steel forgings, depending upon their size, and are drilled and slotted to receive the shanks of the buckets.

The main bearings are of the self-alining, split, babbit lined type and on the larger sizes are provided with oil under a pressure of about 6 lb., the oil being supplied from a reservoir in the base plate and the

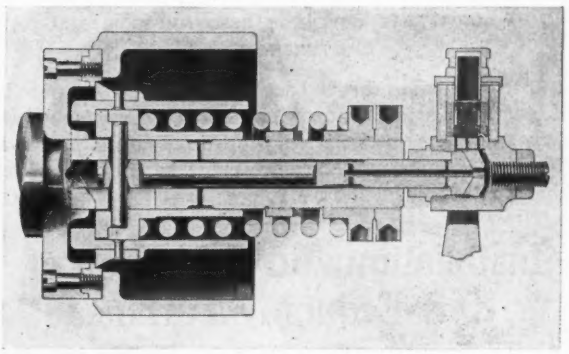


FIG. 5. SECTION OF STANDARD DIRECT ACTING GOVERNOR

required pressure obtained from a rotary pump attached to the lower end of the governor spindle. Before being used, the oil is forced through a nest of water-cooled brass tubes located in the baseplate and after having passed through the bearings is returned to the suction reservoir. On large units a small auxiliary steam turbine driven centrifugal oil pump bolted to the suction reservoir and with the impeller submerged in the oil automatically serves the bearings while the turbine is being parted or stopped.

On machines of the smaller sizes ring oilers are employed.

Leakage of steam along the shaft at steam and exhaust ends and between stages is prevented by floating carbon packing rings held in place by annular spiral springs and stops. The packing at the steam end consists of a number of these rings with two leakage connections taken from selected points between them and piped to the low pressure stages. A low pressure, which reduces the tendency to leak along the shaft, is then maintained on the outer ring. On condensing turbines, one of these leakage pipes is connected to the exhaust and packing so as to form a steam seal and prevent air being drawn through this packing into the condenser.

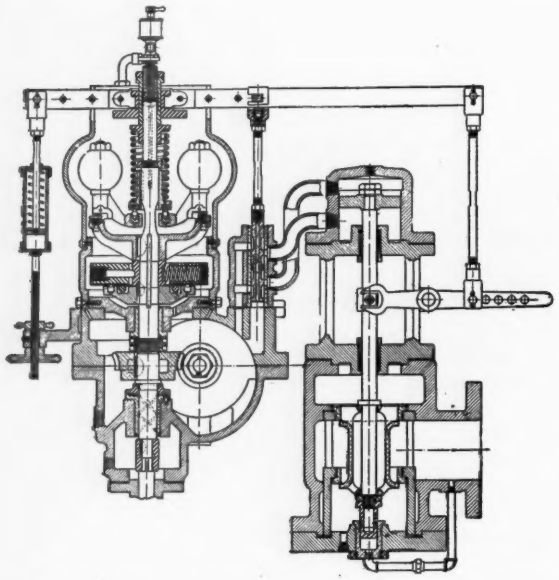


FIG. 6. OIL RELAY GOVERNOR USED ON KERR ECONOMY TURBINES

An adjustable thrust bearing is provided to permit of endwise adjustment of the rotor wherever necessary and to maintain it in proper operating position.

Kerr Economy turbines are made by the Kerr Turbine Co., of Wellsville, N. Y.

Installation and Operation of Kerr-Economy Turbines

THE Kerr-Economy turbine, if properly installed, requires but little care and hardly any adjustments. It might be well to mention here a few points which are considered by the manufacturer as the essentials of a proper installation. These points, while they apply not only to a Kerr unit but to turbines of any type, cannot be too strongly emphasized.

Foundations should, where possible, be isolated from foundations and walls of the building, and made solid rather than hollow. Piping should be supported so as to relieve the turbine casing of all strains due to weight, expansion or unbalanced pressure. A steam strainer should be provided ahead of the turbine throttle to prevent foreign matter from entering through the steam line, and it is desirable to install a separator to free steam from moisture. An expansion joint should be placed in the exhaust line near turbine to relieve the turbine casing. For condensing turbines an atmospheric exhaust valve should be placed between the turbine and condenser, without shut-off valve between the turbine and this exhaust valve. It is desirable to place a valve between the exhaust valve and condenser to allow the unit to operate non-condensing in case of repairs on the condenser.

Drains from turbine throttle valve, separator and from bottom of exhaust end casting may be connected to the condenser when that is below, or to hot well pump or trap when the condenser is above the exhaust end flange.

Every unit should be correctly lined up. Too much stress cannot be laid upon this point. The majority of troubles experienced with turbine units are traceable directly to incorrect alinement at the start or a change in alinement due to steam or exhaust piping pulling the unit out of line. Contrary to common supposition, no coupling has yet been made flexible to such a degree that it will permit of an appreciable error in alinement between the steam turbine and driven machine without imposing such stresses upon both machines as to cause trouble, and even perhaps ruin the shaft in one or the other of the two.

In regard to the operation, the manufacturer makes the following recommendations:

A good lubricating oil, free from acid, and one that will not emulsify, should be used, and it should be kept clean.

The wearing parts of the governor, the knife edges and seats, should be examined at least once every 6 mo.

It is also desirable to dismantle the turbine every 6 mo. and examine the carbon rings, replacing any that may have become damaged; this to improve the steam economy. Outside of this, there is generally no need for dismantling the turbine. This type of turbine is free from bucket erosion because of its low steam velocities, and such troubles as falling off in power are consequently unknown. Also any adjustment of axial position of the rotor is unnecessary. Valves in the sealing pipes of the gland for condensing turbines should be adjusted for great changes in load to prevent an excess of steam from blowing into the turbine or escaping into the engine room. This is especially true in the case of a condensing turbine which must at times operate with full or partial load non-condensing.

Moore Steam Turbines

MOORE steam turbines, which are made by the Moore Steam Turbine Corporation, of Wellsville, N. Y., are built in sizes ranging from 5 to 1000 hp. and in single and multi-stage forms. Speeds of the smaller sizes are limited to 5000 r.p.m. and those of the larger sizes to 3600 r.p.m.

As indicated in the accompanying illustrations, the lagged casings are split horizontally, thus allowing ready access to the interior of the machine. Diaphragms also are divided horizontally with the two halves attached permanently to the upper and lower halves of the casings by a set of special tapered screwed plugs inserted in

the vertical joint while the diaphragm and casing are held rigidly in a fixture provided for that purpose.

The rotor consists of a single pressure, two-velocity stage wheel carrying two rows of buckets followed by two or more single pressure stage wheels mounted on the same shaft. Wheels and shaft are of steel and the buckets are of steel drawn or machined smooth and set into a dovetailed groove in the periphery of the wheel, the form of construction employed allowing the ready removal of the buckets when renewal becomes necessary.

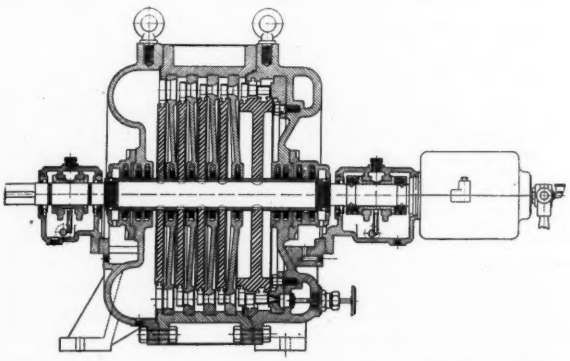


FIG. 1. SECTION THROUGH MOORE MULTI-STAGE TURBINE

In the single-stage turbine and in the first stage of all multi-stage machines, brass nozzles of the diverging expanding type, cast and finished to size are used. Those of the second and following stages are formed by casting sheet metal strips with the diaphragms.

Reversing vanes are used between the two rows of moving buckets of the first stage wheel. These are set into a bored dovetailed groove in the steam end casting and are held securely in place.

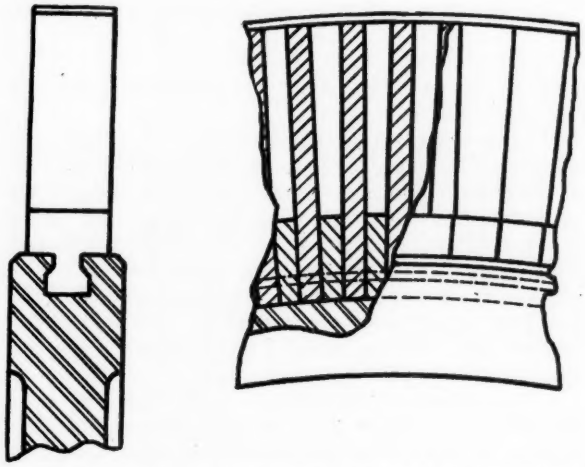


FIG. 3. METHOD OF ATTACHING BUCKETS TO WHEEL OF MOORE TURBINE

Ring or forced lubrication or both forms, depending upon the size of the turbine and operating conditions, are used on the main bearings. End thrust of the rotor shaft is overcome in the smaller machines by providing steel thrust collars so that the endwise position of the rotor may be changed if desired and the clearance between the collars and the ends of the bearing altered to take up excess wear. In the larger machines a Kingsbury thrust bearing is employed.

For ordinary service, a governor of the direct-connected type, Fig. 4, is employed. This is mounted on the end of the turbine shaft and actuates a steam regulating valve through a lever connection. Where, how-

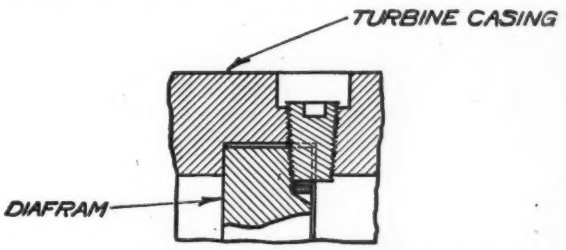


FIG. 2. METHOD OF ATTACHING DIAPHRAGM TO CASING OF MOORE STEAM TURBINE

ever, extremely close regulation is required, as in the case of electric generating units, a highly sensitive oil relay governor is generally provided. In this an oil relay control is operated by a Jahns governor in which

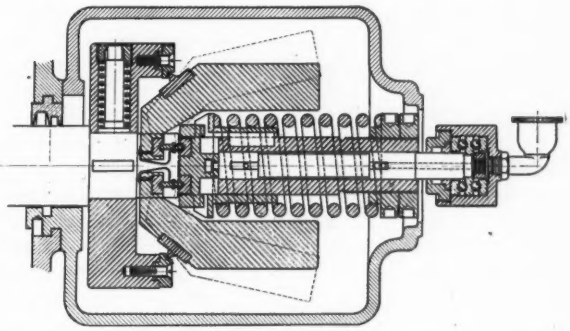


FIG. 4. SECTION OF STANDARD GOVERNOR USED ON MOORE TURBINE

the weights are so arranged that the spring pressure comes directly against the weights, so that the pressure exerted by the weight does not enter as a factor in the friction of any of the moving parts.

Ridgway Turbines

TURBINES manufactured by the Ridgway Dynamo & Engine Co. are designed under the patents of A. Rateau and C. H. Smoot. They are of the impulse type, in which the energy of the steam is changed, by expansion, into kinetic energy, which in turn is changed into work as this energy is absorbed by the moving buckets against which the steam is directed. The steam enters through a set of stationary nozzles in which it expands to a lower pressure with a corresponding increase in velocity. As the steam issues from the nozzles, it strikes the entrance edge of the first row of buckets and in passing through them its direction is changed, giving up energy to the buckets and driving them forward.

The steam next passes through the second set of stationary nozzles, which are of greater area than the first in order to give room for the greater volume of the

steam. Here the steam again expands and enters the second row of buckets at the same velocity as when entering the first. The steam passes through the second row of buckets as it did the first, and the process is repeated through several stages until it reaches the exhaust outlet.

In each stage only one row of revolving buckets is employed and the pressure between any two sets of stationary nozzles is the same on either side of the revolving buckets. For these reasons, there is no necessity for close running clearances between the buckets and the nozzles or casing. In small machines, the minimum

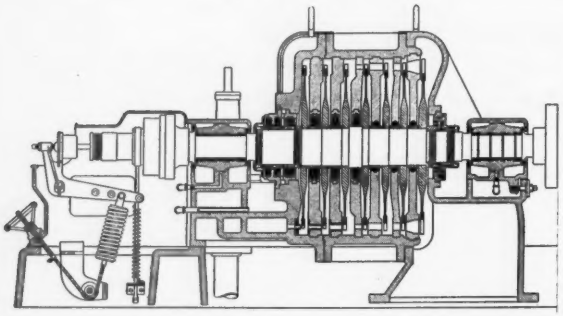


FIG. 1. SECTION OF RIDGWAY HIGH PRESSURE TURBINE

clearances are 3/16 in. axially and ½ in. radially; in large turbines, clearances are as much as ⅜ and 1 in. respectively.

Mounted upon the shaft are the wheels, which carry at their periphery the revolving buckets; this assembly constitutes the rotor of the machine. The shaft is forged from high carbon steel of such a diameter that its normal speed is below its critical speed. On one of the two journals are turned a few thrust rings, serving to keep the wheels and nozzles in proper relation and to oppose

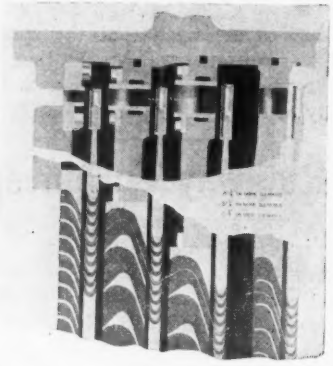


FIG. 2. SECTION OF RIDGWAY NOZZLES, BUCKETS AND WHEELS

the thrust of the governor spring. The wheels are machined from disks of flange steel and are keyed on the shaft. The buckets are machined from solid bars of a special alloy and are made in two types, depending upon their size. The smaller buckets are secured by rivets through their shanks, so placed as to retain the maximum section of the shank. Large buckets are made with bulb ends which are driven into slots in the periphery of the wheel and peened solidly into place. Each bucket of either type carries its own shroud and, when assembled on the wheel, is in rigid contact, at the outer end, with the adjacent buckets. By reason of this construction, all buckets are supported by the others in the event of foreign substances getting past the first nozzle; vibration of individual buckets and consequent breaking is prevented; any bucket may be removed or replaced without disturbing adjacent buckets.

Being split horizontally, the upper halves of the casing and heads may be easily lifted off in case it is necessary to open the turbine. The diaphragms are also split and the top halves are fastened to the top of the casings and lift with it. Steam and exhaust connections are made to the lower half and need not be disturbed when the turbine is opened.

The nozzle construction in these turbines varies in the different diaphragms, depending upon the area required. Nozzles with small area are solid castings of special alloy, machined and accurately shaped. Such nozzles are used in the first stages and are bolted in place. In later stages, the blades forming the nozzles are cast in place in the diaphragm and extend all or part way around the periphery of the diaphragm.

Packing for high-pressure boxes and diaphragms is made of carbon blocks and for the low-pressure boxes

FIG. 3. FRONT AND BACK OF HALF DIAPHRAGM WITH CAST NOZZLES

a water impeller is used, which acts similarly to that of a centrifugal pump.

The governor is of the throttling type with the valves directly connected to the flyball governor mounted on the end of the main turbine shaft. The weights are carried on tool-steel knife edges and the spring thrust is taken on tool-steel cut joints so that no lubrication is required. The main governor thrust bearing is provided with two ball bearings operating in oil. The auxiliary spring connected to the bell crank underneath the governor provides speed adjustment while the turbine is in operation.

Various types of valves are used depending upon the conditions of service. For high pressure, the main valve is usually of the double balanced poppet type feeding an arc of nozzles sufficient to carry full load. A separate valve of the piston type, mounted on the same stem, supplies a separate set of nozzles for carrying overloads, but does not open until the main valve reaches full load position. An additional hand-operated valve is sometimes used in noncondensing service, which bypasses the first and sometimes the second stage, but keeps the turbine under full control of the governor. For emergency use, an automatic and hand-operated over-speed trip is provided.

Lubrication of the main bearings is accomplished by a gravity pressure system in which the oil is purified and cooled before reuse.

Ridgway turbines are also designed to use exhaust steam at about atmospheric pressure. These differ but little in general design and appearance from the high pressure machines, the difference being that the valves and nozzles are larger to accommodate the greater volume of steam required. These low-pressure turbines are not equipped with either automatic or hand-operated

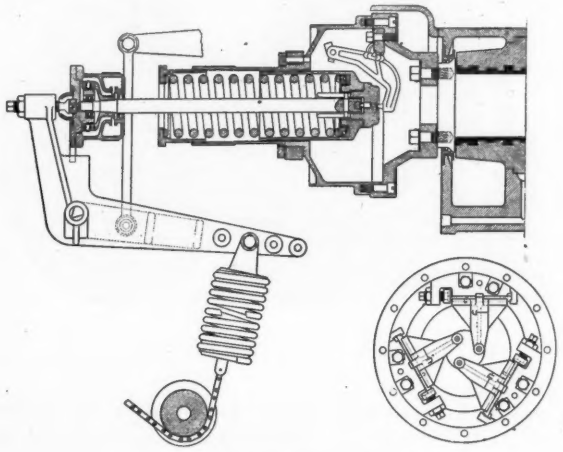


FIG. 4. DETAIL OF GOVERNOR

overload valves, as they are designed to be used where there is an ample supply of low pressure steam. Should it be necessary to provide for an occasional failure of the low-pressure steam, high-pressure steam may be supplied through a reducing valve, adjusted to open only when the pressure of the low-pressure steam falls below atmosphere.

What is known as a mixed-pressure turbine is virtually a low-pressure turbine with the addition of one

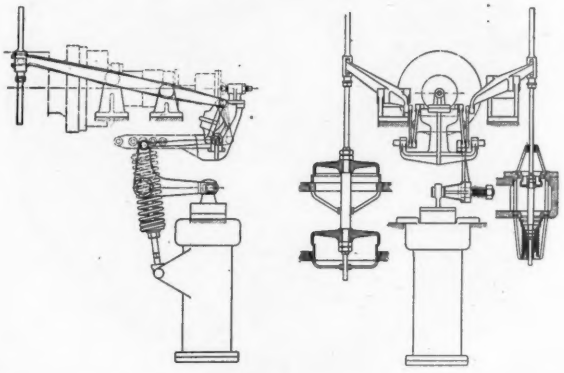


FIG. 5. DETAIL OF MIXED-PRESSURE GOVERNOR AND VALVES

or more high-pressure stages, which exhaust into the first low-pressure stage. This type is designed for use where there is a frequent failure in the low-pressure steam supply, or where it is desired to use all the low-pressure steam available, making up the deficiency with high-pressure steam. An outline of the governor and valve construction is shown in Fig. 5. The large balanced valves admit low-pressure steam to the low-pressure stages and the small balanced valves admit high-pressure steam to the high-pressure stages. The cylinder under the governor is connected to the low-pressure steam supply, which, if ample to carry the load, will raise the plunger in the cylinder, and by means of the small bell cranks and connecting links, will throw the low-pressure valve into action under control of the governor. If the supply of low-pressure steam is not sufficient, the plunger will descend, throwing the low-pressure valve out of action and bringing the high-pressure valve under control of the governor, or both sets of valves will take an intermediate position, using all the available low-pressure steam and making up the deficiency with high-pressure steam.

The Steam Motor

THE Steam Motor made by the Steam Motors Co. of Springfield, Mass., is a small steam turbine of the single-stage, multi-velocity type, as shown in Fig. 1. It is mounted with one bearing only for direct positive connections to pumps, generators or other driven machinery in order to secure compactness, light weight and certainty of alinement. By removing one bearing from the driven apparatus and substituting the bearing supplied with the Steam Motor, a complete unit results, having but two bearings for the whole equipment.

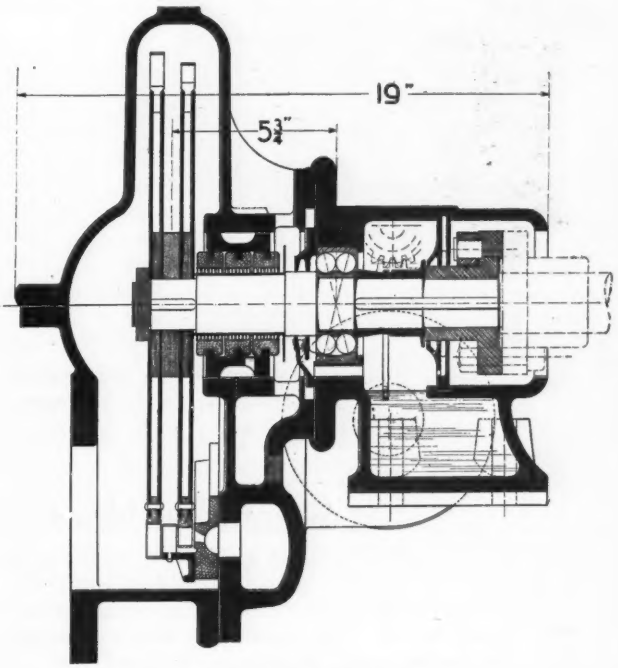


FIG. 1. SECTION OF STEAM MOTOR

At the point where the shaft leaves the casing the gland or stuffing box is of double-labyrinth type. The wheel is of plate construction mounted on a substantial hub keyed to the shaft and held by a suitable locking nut. Buckets are of drop forgings and are riveted between the disks.

The machine can be fitted either with emergency or constant-speed governor. Where emergency governor only is required, as for boiler feed pump work, this gov-

ernor is in duplicate. In other words, two separately-adjusted weights are fitted so as to insure a positive shutdown in the case of runaway. Figure 2 shows the appearance of the large units with upper half of casing lifted and upper half of pillow block opened showing the arrangement of wheels, packing gland and bearing.

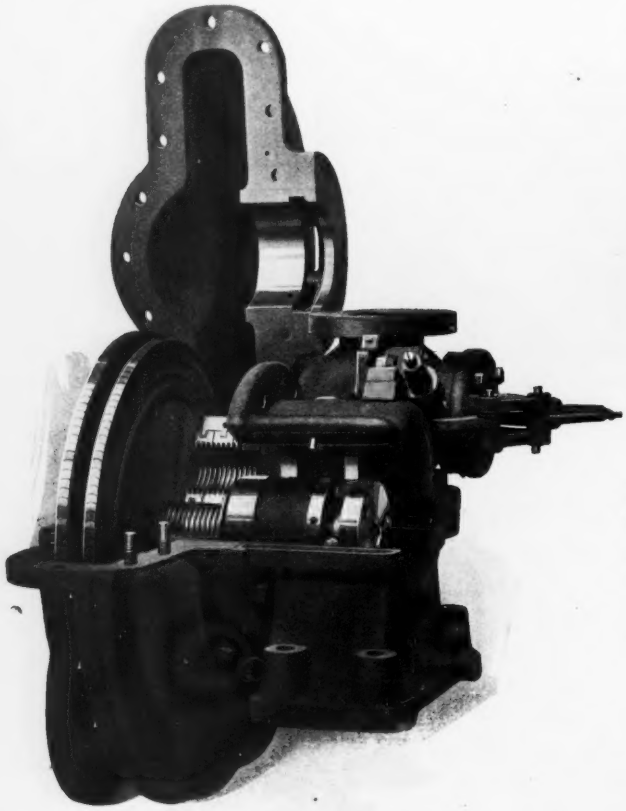


FIG. 2. STEAM MOTOR OPENED UP FOR INSPECTION

In addition to the type shown in the illustrations, the Steam Motors Co. has developed for direct connection to small pumps and blowers a series of sizes as small as 1 hp.

Sturtevant Steam Turbines

BEING designed to use steam on the impulse principle, the detail of special interest in Sturtevant turbines is the steam nozzle and reversing buckets, which are made of Tobin bronze cast together in segments, accurately milled, ground and polished.

These nozzles are designed to expand the steam from boiler pressure down to that of exhaust before reaching the rotor buckets. At the resultant high velocity, the steam impinges against the rotor buckets, imparting the impulse to the rotor and leaves the buckets in the reverse direction. As it leaves the rotor, the steam enters the reversing buckets which again reverse the direction of the steam and drive it back into the rotor. The steam enters and leaves the rotor several times before its kinetic energy is absorbed and its velocity drops to that of the rotor. It then passes out into the exhaust.

The stationary reversing buckets are semi-circular in form, and are set at angles which have been found to give the most efficient results. These nozzles are bolted to the inside of the casing known as the steam head, as shown in Fig. 1, and receive steam from the annular steam chamber within the casing.

By means of hand operated valves on the exterior of the casing, the nozzles can be separately closed. Thus

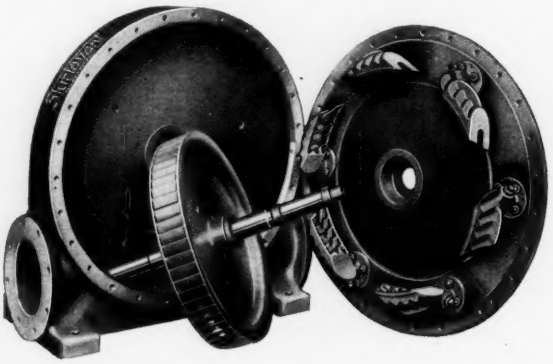


FIG. 1. PARTS OF THE STURTEVANT TURBINE

by closing one or more of the nozzle valves, the turbine will run economically on continuous light loads, the governor taking care of fluctuations in the same manner as when all valves are open.

The rotor or bucket wheel is a solid forging of open-hearth steel with buckets milled into the rim. The

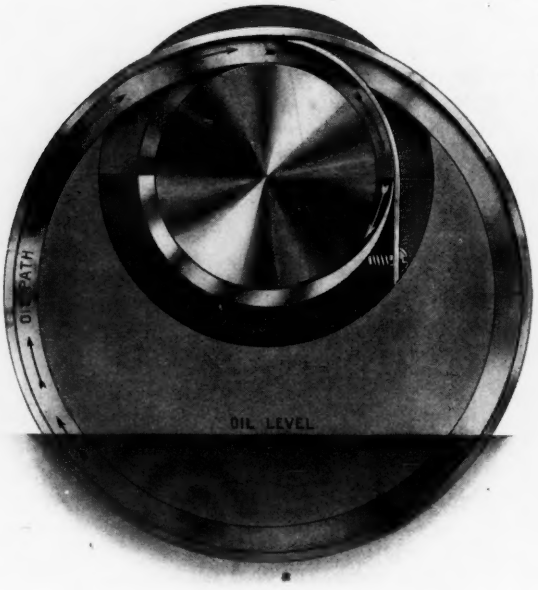


FIG. 2. CHANNEL-SCOOP OILING DEVICE

buckets are semi-circular in form and reverse the direction of the steam, driving it back into a set of stationary buckets of similar design. The rotor buckets are protected by the rim of the wheel which is raised above them and the construction of the wheel is such that if by any chance they should come in contact, none of the buckets

would be injured. The rotors are given static and dynamic balance at the speed at which they are designed to operate resulting in smooth, quiet running and eliminating the chance of the shaft springing enough to throw the rotor in contact with the reversing buckets.

Nickel steel is used in the shaft which is turned and ground to size to eliminate all chance of spring or uneven balance. Two threaded steel thrust collars on the governor end make it possible to adjust the rotor to any changes in the register of the rotor. Metallic packing is used around the shaft where it passes through the

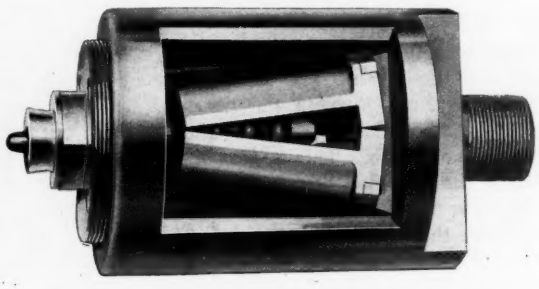


FIG. 3. GOVERNOR OF STURTEVANT TURBINE

casing being held in a floating stuffing box and is required to carry only the difference between exhaust and atmospheric pressure. For use with a condenser the packing has a water sealing groove.

In Fig 2 is shown a cross section of a main bearing to illustrate the method of lubrication which is by means of a channel oiling ring. The oil is taken from the oil pocket by the rapidly revolving channel-shaped oiling

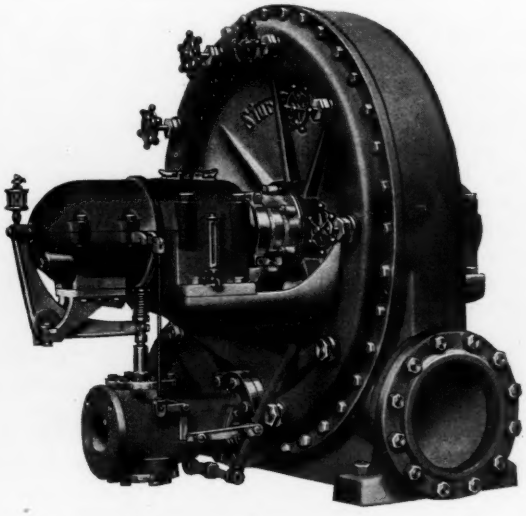


FIG. 5. ASSEMBLED VIEW OF STURTEVANT TURBINE

ring and is held within the ring by centrifugal force until it is diverted by the bearing by means of a scoop.

Speed is regulated by means of a balanced throttle valve operated by a governor of the centrifugal type. Two revolving weights are made, by centrifugal force, to act against a hardened steel spindle. Movements of this spindle are transmitted by a single lever with knife-edge bearing to the regulating throttle valve controlling the opening and closing of the throttle valve.

In addition to the main governor, there is a separate emergency governor which is set at a predetermined maximum speed. If for any reason, the main governor is unable to control the speed of the machine, the sepa-

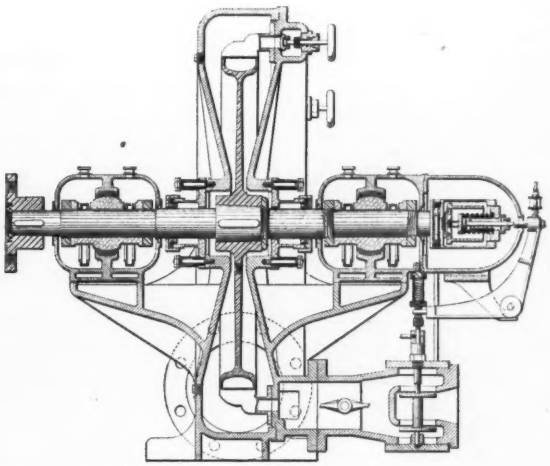


FIG. 4. SECTION OF STURTEVANT TURBINE

rate emergency valve is instantly and automatically closed by the emergency governor working through a separate mechanism of its own.

Instructions for Installing and Operating Sturtevant Turbines

By O. D. H. Bentley

ALL Sturtevant turbines are thoroughly tested before shipment, and should be in perfect condition when received by the customer, unless the adjustments have been deranged or tampered with in transit. It is a good plan to look the turbine over carefully in order to determine that no parts are broken or damaged.

ALINEMENT

The most important alinement relates to the so-called flexible coupling, between the turbine and the driven apparatus. It is very important to make sure that the face and sides of the couplings run parallel and true with each other; in other words, that the two shafts are in perfect alinement.

The preliminary alinement is usually made when the apparatus is cold and before the piping is connected. It must be borne in mind that the turbine rises when the steam is turned on, therefore, the following allowances should be made to compensate for this change; that is to say, the turbine coupling should be lower than the driven coupling, per the following table:

Size Turbine	Change in Turbine Height
A	0.006 In.
B	0.008 In.
C	0.010 In.
D	0.012 In.
E	0.014 In.

The final alinement should be made after the turbine is connected up with the piping and ready to be put in operation and, in case the turbine is grouted in, it should not be done until after this final alinement is checked.

The turbine should be so piped that no strains are imposed upon the casing, and all piping must be properly supported externally from the turbine. Before connecting the piping, it should be thoroughly blown out with high-pressure steam, and in some installations, where scale and sediment are apt to continue to come through the pipe line, it is desirable to install a steam strainer just before the turbine valve.

The final alinement of the coupling should show that the coupling is not out more than 0.002 or 0.003 in., otherwise, vibration will result and the rubber bushings will wear rapidly.

Special attention is called to the above facts, for engineers have sometimes reasoned that inasmuch as the apparatus comes to them self-contained and mounted upon a sub-base, and also because there is a semi-flexible coupling between the turbine and the driven apparatus, no further attention should be required from them, before putting the apparatus into service. Experience has shown that it is not safe to depend upon a sub-base to hold high speed apparatus in alinement, nor is it possible for a flexible coupling to take care of any appreciable mis-alinement at high speed.

Starting

When starting the turbine, it is advisable to bring up the speed gradually until the turbine is under control of the governor. In this way, the operation of the governor can be checked. In shutting down the turbine, it is advisable to trip the emergency valve in order to make sure that it is functioning properly.

Lubrication

The two main bearings and the governor pin are the only places requiring lubrication on the turbine. The bearings should be looked after carefully and replenished with oil as the occasion demands and thoroughly flushed out periodically with kerosene, and fresh oil supplied. The lubrication at the governor point should be given careful attention, otherwise, wear will take place and the governor operation will be impaired.

All first class oil companies furnish an oil especially made for turbines, usually called "Turbine Oil," although many of the oil companies have a trade name. A good mineral oil that shows a viscosity of 200 to 300 sec. at 100 deg. F. based on the Saybolt Universal Test, will give satisfactory results. Avoid using cheap oil for it is expensive in the long run. Good oil is cheaper than bearings.

Packing

Sturtevant turbines are furnished with floating stuffing boxes and should be given the same attention as an ordinary stuffing box on the piston rod of a reciprocating engine; that is, the gland should be followed up or tightened from time to time, in order to prevent leakage, although a slight leak, however, is permissible and helps to lubricate the gland, but a leak that blows steam should not be tolerated.

When the gland has been tightened as far as it will go, another ring of packing should be added. It should never be necessary to take out the old packing, simply add rings as required.

Clearance

The radial clearance and longitudinal alinement can be determined through the inspection plugs opposite the nozzles. The radial clearance should be divided equally between the various nozzles and should not be less than 1/32 nor more than 1/16 of an inch. The radial clearance can be changed by shimming under the main bearing casing.

The longitudinal alinement or registration of the rotor and nozzles can also be determined through the inspection plugs opposite the nozzles. The alinement is correct, when the edge of the wheel is flush with the edge of the nozzle. This adjustment can be made by shifting the threaded thrust collars on the bearing next to the governor. Care should be taken, however, to lock these thrust collars securely. The shaft should be drilled for the points of the set screws, in order positively to hold the collars in their locked positions.

Nozzle Valves

Sturtevant turbines are equipped with nozzle valves on some of the nozzles, which makes it possible to get exceptionally high efficiency under light loads. The idea is to operate the turbine with as few nozzles open as possible, corresponding to the load.

Governor

The speed of the turbine can be changed by adjusting the threaded nut in the end of the governor. The speed will be increased by screwing in, or lowered by backing out. The valve should be adjusted when the turbine is at rest and with the throttle closed. A 3/16-in. block should be imposed between the governor pin and the bell crank. This has the effect of lifting the valve. If the throttle is now opened, the gage on the steam chest should show from 10 to 25 lb. pressure, in other words, the valve should be slightly open. This setting will insure the valve closing sufficiently to prevent the turbine from running away at light loads. The governor has a travel of ¼ in. from one extreme to the other.

The function of the governor is to close or lift up the balanced steam valve. The external spring connected to the bell crank is depended upon to open the governor valve; therefore, if the governor valve refuses to open, increased tension should be put upon this external spring, but no more pressure should be used than necessary, for it imposes a considerable pressure on the governor spindle and makes it all the more important to look after the lubrication at this particular point.

The emergency governor can be adjusted by screwing in or out on the plug opposite the opening, through which the plunger projects. A very slight change in this adjusting screw will make a perceptible difference in the speed at which this governor operates. Care should be taken to prick-punch the adjusting screw after final adjustment is located. The emergency governor can be tested by removing the bell crank from the main governor and gradually bringing the speed up until the

emergency trips. Attention is called to the fact that the threaded bushing through which the governor projects should not be altered, for this determines the amount that the governor plunger comes out. By inserting a small, stiff wire through the hole in the center of the adjusting screw, the governor pin can be forced out, but this should not extend more than 3/16 or 1/4 in. in the maximum out position.

Bearings

In adjusting the bearings on the Type 5 Sturtevant turbines, attention is called to the fact that there are shims, not only between the bearing linings, but also between the bearing casings. Therefore, it is important not only to adjust the bearing linings properly on the shaft, but also to adjust the bearing casings snugly in order that the spherical seats will grip the ball on the bearing linings.

On the Type 6 Sturtevant turbines there is an adjusting screw used in the bearing casing, which is simply tightened down, causing the spherical seat to grip the linings. This adjusting screw does away with the shims used in the Type 5 turbines.

The section through the Sturtevant turbine shown in this issue, clearly illustrates the construction of this turbine and shows to an engineer, the points of adjustment and the order in which the turbine should be taken apart, if it is ever necessary to take it down, although this is very seldom required, as practically every adjustment can be made without dis-assembling the turbine.

The Terry Turbine

THESE turbines are made in two types, the noncondensing with one wheel of single pressure, multivelocity stage type; the condensing with a composite element which expands the low-pressure steam through a series of multi-pressure, single velocity stages to condenser pressure.

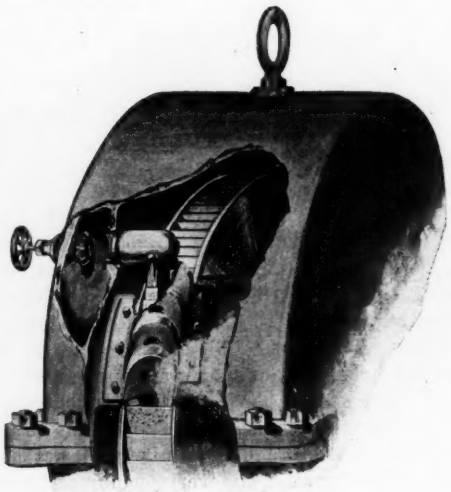


FIG. 1. ACTION OF STEAM IN TERRY TURBINE

The rotor of the noncondensing turbine is made from a single forging of special steel having semicircular buckets milled from the solid metal of the rim, so as to leave protecting rings at the sides. Several nozzles are mounted about the periphery to discharge into the buckets at one side, the steam flowing tangentially, being reversed by the bucket and passing to a reversing chamber which returns it to the wheel buckets, Fig. 1. This is repeated until all available energy has been taken up by the wheel.

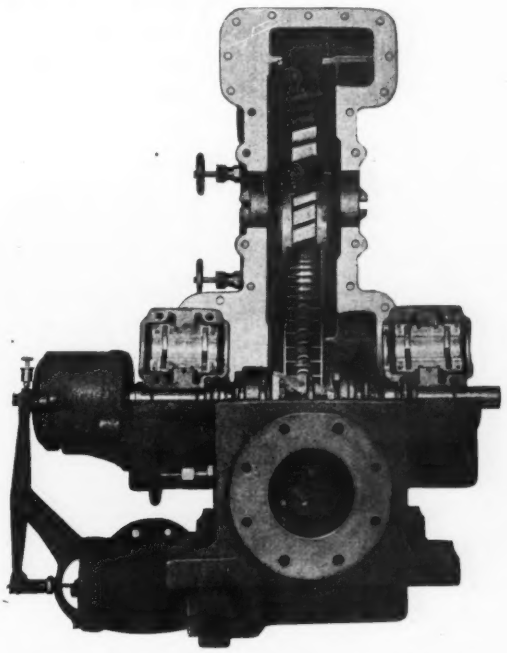


FIG. 2. TERRY TURBINE WITH CASING OPENED

All energy is given up on the curved surfaces at the back of the buckets, the blades serving only to split the steam jet, hence close blade clearance is not sought and blade wear has little effect.

Nozzles are made separate from the reversing chambers, so that they can easily be changed to suit different operating conditions of pressure and superheat. Each nozzle is controlled independently by a hand valve, so that the best number to carry any required load may be employed, so as to give best economy at partial loads and a minimum of throttling by the governor. Arrangement of the parts is clearly shown in Fig. 2, where the upper half of the casing with attached nozzles and reversing chambers is raised.

The wheel is secured to a short, sturdy shaft by means of a key and lock nuts, the shaft passing through bushings in the casing and resting in ring oiled bearings in the pedestals. As only exhaust pressure is carried in the casing, which really serves only as a means of collecting exhaust steam and carrying it to the exhaust pipe, there is little chance for leakage.

Figure 3 shows the arrangement of parts and the location of the governor on the end of the main shaft. By means of a rocker arm, this works directly on the stem of the balanced throttle valve giving positive action.

In the "composite design," shown in Fig. 4, steam from the high-pressure casing is passed to the far end of the low-pressure element, surrounding that element and the exhaust chamber, thus steam jacketing the vacuum spaces and preventing inleak of air.

Internal diaphragms are horizontally split so that each one can be removed from the casing without disturbing the others. The shaft, of large diameter, is stepped for the wheel hubs, and labyrinth rings are used to prevent leakage along the shaft and between stages.

For low-pressure work only, taking steam from a reciprocating engine, the Terry low-pressure element may be used; but, if the high-pressure wheel is installed as a reserve, the unit can be used in emergency as a high-pressure condensing turbine, or if the exhaust steam is not sufficient to drive the low-pressure element at full

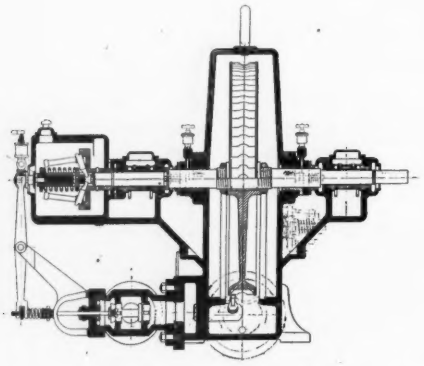


FIG. 3. SECTION OF TERRY TURBINE SHOWING GOVERNOR ACTION

capacity, some steam may be passed through the high-pressure wheel, giving added power.

In such case, high and low-pressure supplies are controlled by individual governor valves, the most satisfactory regulation being obtained by a mechanism known as the Rateau mixed pressure governing system, which automatically compensates for changes in pressure in the low-pressure steam line and prevents such changes from disturbing the regulation of the turbine. Low-pressure

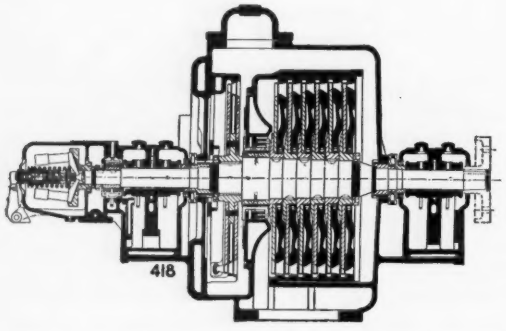


FIG. 4. SECTION OF TERRY RETURN FLOW DESIGN

connection for the mixed-pressure arrangement is made through the opening covered by a dome plate in Fig. 4. For large units, where the low-pressure inlet valve is of considerable size, an oil relay governor is used. Bleeder connections can also be provided.

The Terry turbine is built in sizes of 5 to 1000 hp. and runs at speeds which permit of direct connection to generators, fans, blowers and pumps.

Installation and Care of Terry Turbines

By Maynard D. Church

SUCCESS of any turbine depends largely on its proper installation and operation. Complete instructions are furnished with each Terry turbine and additional copies will be supplied the engineers on receipt of the turbine number and name of the plant. The following brief suggestions will give an idea of some points to be observed.

Foundations

Turbines do not require such massive foundations as reciprocating machinery, but the foundation must be sufficiently heavy to afford permanent rigid support. Any baseplate will deflect to some extent if not properly supported, causing misalinement.

Concrete foundations resting on the earth are best when practicable.

When a turbine driven unit is supported by steel work it is well to provide local stiffness by extra staying and bracing, or by the addition of a substantial concrete mat either above or below the floor.

In case turbines are installed without a base plate, suitable cast-iron or steel sole plates should be provided under the feet of the turbine and the driven machines. These sole plates should be grouted into place and the feet of the machine bolted and doweled to them.

Alinement

Correct alinement is essential to successful operation of any high-speed machine. Final alinement must be made at installation. Every baseplate is elastic, no matter how deep or heavy. Terry baseplates are unusually liberal, but they can spring to some extent.

Every unit assembled at the works is accurately lined up and each part doweled in place. During alinement the baseplate rests on a number of supports well distributed around the edge, and the planed pads are kept level. When installing the unit it is only necessary to support the base in a similar manner, preferably on iron wedges, and by adjusting these wedges to bring the unit into correct alinement.

When alinement is correct the faces of the flexible coupling will be parallel with each other when tested with "feelers" or a wedge. The turned rims of the couplings will also be in alinement as shown by a straight edge. To allow for heat expansion when lining up cold, the turbine should be set from 0.006 to 0.012 low, depending on the size of the turbine and the temperature of the exhaust steam. The alinement should be checked up with the turbine hot and adjusted if necessary. It is important to remember flexible couplings will not fully compensate for misalinement. Every baseplate is to some extent elastic, therefore every unit must be lined up in place. This applies to every combination of two or more machines on a base plate, whether turbine machinery or any other kind.

Piping

Pipe sizes named by the turbine builder are for average conditions. When lines are long or contain

many fittings, use steam pipes large enough to insure rated steam pressure at the throttle, and exhaust pipes ample to insure against loss of vacuum or building up excessive back pressure.

All piping should be so arranged that neither its expansion and contraction nor its weight can put heavy strains on the turbine. This applies especially to exhaust pipes. Piping on centrifugal pumps should be considered in the same manner. Exhaust connections on Terry turbines are low and close to the feet so trouble from pipe strains is minimized, but there is always danger of casing distortion or misalinement from badly arranged

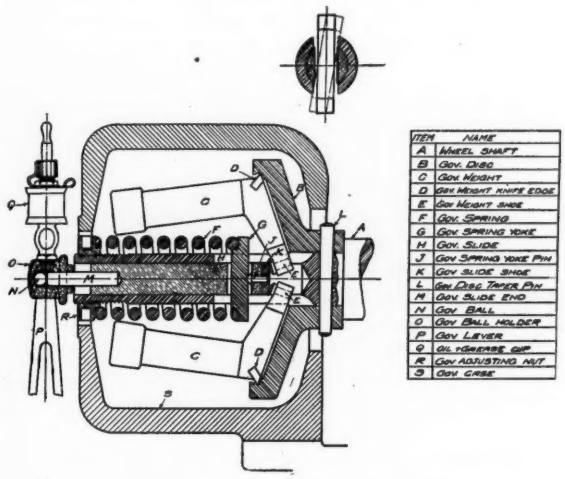


FIG. 1. SECTION OF TERRY GOVERNOR

piping. Exhaust pipes 8 in. in diameter or larger should be provided with expansion joints close to the turbine, and smaller pipes also, if otherwise rigid.

All steam piping should be thoroughly blown out with steam before connecting to the turbine. It is also desirable to put a suitable strainer close to the turbine inlet.

To insure dry steam, all steam pipes should be lagged, and a separator with trap, near the turbine, will usually pay. All steam pipes should be taken from the top of the header when possible. Even superheated steam may contain moisture, and moisture is detrimental to both the economy and the life of any steam turbine.

Starting a New Turbine

First examine carefully to make sure no parts have been damaged in shipment or handling. Clean all parts. Flush bearings, oil reservoirs and gear cases with clean kerosene, and fill with lubricant as recommended under subject of Bearings. If cooling oils or jackets are provided, turn the water on and look for leaks, especially leaks into the oil spaces. Turn the unit by hand to be sure it is free.

On the first run, drain the turbine, start slowly and be sure that oil rings or oil pumps deliver plenty of oil. Test speed frequently with a tachometer or speed counter. If an oil relay governor is provided, see that the oil pressure picks up the governor valve well before full speed is reached. Watch all bearings carefully, feeling the temperature directly on the bearing shell if possible, using clean fingers. Heating when starting up may be due to grit or misalinement.

As the turbine approaches full speed, observe the action of the governor. When the governor is in control, open the throttle and the turbine is ready for load.

Routine Operation

When in regular use, the turbine can be started more quickly. With proper lubrication and an occasional inspection, it is only necessary to open the throttle on small Terry turbines, provided the driven machine does not require special care.

For best economy, keep as many hand valves closed as possible.

Governor

The Terry governor, Fig. 1, is of a simple powerful type and requires little attention beyond occasional lubrication of the "Ball Holder," which is the little thrust block through which the governor transmits its motion to the governor valve. Most governor troubles can be avoided by keeping all parts of the governor mechanism clean and in good adjustment. Speed fluctuation may be caused by dirt or bruises in some part. Overspeeding is usually due to the governor valve being worn or improperly adjusted.

Governor Valves and Valve Setting

All Terry governor valves are of the double seated balanced type, Fig. 2.

Valves should be so adjusted that the maximum valve opening is 1/8 the valve diameter. This can be measured when the turbine is not running. First push in on the valve stem until the valve is seated, and make a mark on the stem opposite the inner edge of the governor valve bonnet. Then pull out on the stem as far as the governor mechanism will allow, and make another

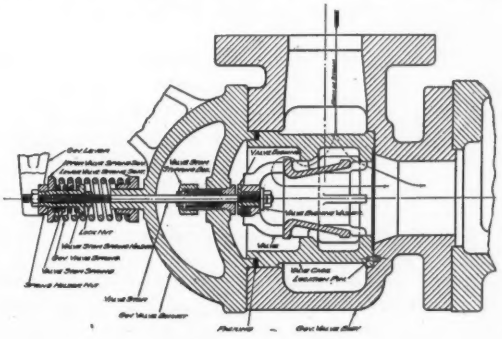


FIG. 2. TERRY GOVERNOR VALVE

mark. The distance between marks will give the valve opening. This opening will be increased by the wear of any part of the governor mechanism, and should be occasionally tried and corrected if necessary by adjustment of the nuts on the stem or guide rods.

Small valves are at times set with a greater opening than standard, but in no case must the valve opening be greater than 7/16 in. on an 18 or 24-in. Terry turbine, or 5/8 in. on a 36-in. turbine or a Type GF. Valves controlled by oil relay governors are an exception to this rule.

Valve Stem Packing

The valve stem stuffing box should be packed with Palmetto packing or with asbestos wicking soaked in graphite and cylinder oil. Too tight packing or hard packing on a rough stem may cause sticking and possible "hunting" of the governor.

Detection of Leak

Small valve leaks can be easily repaired, but if not promptly stopped will rapidly get worse, become difficult to stop and be a possible source of serious trouble.

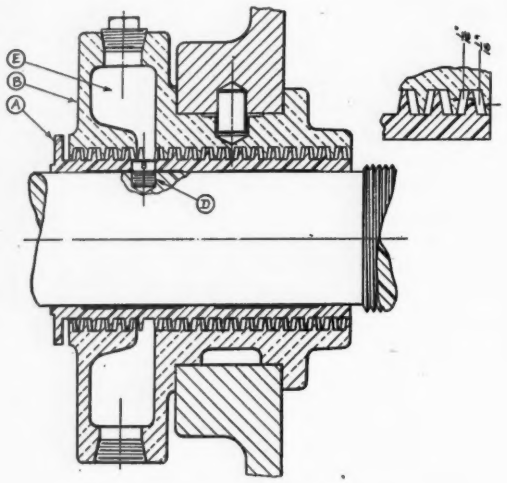


FIG. 3. LABYRINTH TYPE GLAND

An easy way to test valve tightness is to close the valve by hand while the turbine is running. If the speed quickly drops, the valve is tight. Some indications of a valve leak are:

1. Speed runs up at light load, with valve apparently closed.
2. Thrust bearing or ball holder heats or shows wear. This is because the governor presses very hard on the valve in an effort to shut off steam.
3. Turbine is shut down by the emergency governor on light load. This may be caused by other things, but the valve should always be tried.

Remedies for Valve Leaks

If a valve leaks, the first thing to do, of course, is to stop the leak by regrinding or reseating the valve. If the valve has apparently not given as long service as should be expected, the cause of rapid wear should be found and removed. Some common causes are:

1. Turbine runs much of the time at light load. This will always increase valve wear. Secure a smaller valve if your load conditions will allow. Write the makers fully as to the heaviest load you must carry, the ring pressure available at the throttle and actual exhaust pressure, and they will advise you whether it is practicable to use a smaller valve. If a smaller valve is used, it will run further open at light loads, reducing the wiredrawing effect.

2. Wet Steam. Wet steam is destructive to all valves which are required to throttle steam. It has been well established that there may be water even in superheated steam pipes, particularly towards the ends of the lines, and at times of light load. When valves show rapid wear it is well to make sure that steam is dry at the turbine throttle. Install a separator close to the turbine. Provide suitable pipe coverings. Be sure that steam headers are properly drained and trapped and that steam is taken from the top of the headers. Even if superheated steam is supplied, calorimeter readings near the turbine throttle for a few days may show moisture present at times. For especially severe conditions, a valve and seats of monel metal or similar material can be provided at an extra cost. While high in first cost, valves of special metals are most economical where conditions are such as to cause rapid wear.

If a valve is reground sufficiently often to keep deep scores from forming, the life of the valve will be much greater.

Emergency Governors

Terry emergency governors are adjusted to operate at 7 to 10 per cent above rated speed. They trip a butterfly valve in the main steam line and on large condensing turbines operate a vacuum breaker as well.

Every emergency governor should be tested at regular times to insure its being free to act when it is needed. This may be easily done by tripping by hand.

Bearings and Lubrication

All Terry bearings are ring oiling, using two rings. The rings pick up an excess of oil and the bearings are so grooved that the oil can easily flow through the bearing, thus cooling the journal.

Forced oiling or water cooling are supplied in addition at an extra cost where conditions require.

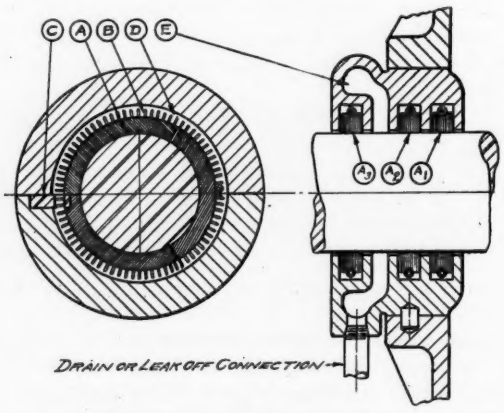


FIG. 4. CARBON-RING GLAND

A good grade of pure mineral oil must be used. It must be of such quality that it will readily and quickly separate from water. If the oil is such that it can form an emulsion with water, considerable damage may result from small leakage of glands or water cooling chambers, or from sweating of water cooled surfaces. If the water separates freely, it may be drawn off occasionally from the bottom of the oil reservoirs, if necessary. The oil must be free from acid, alkali or any foreign matter in suspension.

The viscosity of the oil used in any case must be suitable for the service. We are listing below oils recommended by several refiners for turbine work, in three ranges of viscosity roughly classified as Light, Medium and Heavy. The approximate viscosity of each oil is given with its trade name. All viscosities are in seconds at 100 deg. F. by Saybolt Universal Viscosimeter. These oils are tabulated for the convenience of the turbine user as being standard brands. If any oil named is found unsatisfactory for the purposes stated, please advise us for our information.

MAKER	LIGHT	MEDIUM	HEAVY
Vacuum Oil Co., Rochester, N. Y.	DTE Light 130-140	DTE Heavy 245-255	DTE Extra Heavy-480
Standard Oil Co. of Indiana, 72 W. Adams St., Chicago, Ill.	Superla Turbine 135	Standolind Turbine 345	
The Texas Co., 17 Battery Place, New York, N. Y.	Cetus 180-200	Alcaid 300-325	Algol 500
Atlantic Refining Co., 3144 Passyunk Ave., Philadelphia, Pa.	Atlantic Light Turbine 135 Atlantic Medium Turbine 195	Atlantic Heavy Turbine-280	
Gulf Refining Co., Frick Bldg. Annex, Pittsburgh, Pa.	Paramount Turbine Medium-200	Paramount Turbine Heavy-350	

(1) Light oil, viscosity 130 to 200 sec., is best for turbines without reduction gears, either ring or forced feed oiling.

(2) Medium oil, viscosity 200 to 350 sec., is used for turbines with reduction gears and either ring or forced oiling.

It is better than a light oil for turbines subject to vibration either from within or from an external source. It will also allow slightly greater bearing clearances. Bearings may run a few degrees warmer with heavy oil than with the lighter grades.

(3) Heavy oil, viscosity 350 to 500 sec., is useful in cases of bad vibration or of bad gears heavily loaded or causing noise. Many times gears can be successfully operated with heavy oil which would be noisy or show rapid wear with lighter oil. Heavy oil works well in turbine bearings except in places where exposure to cold may sometimes make the oil too sluggish. This applies especially to forced oiling units. When using heavy oil, more attention must be given to the oil when starting, to be sure that all rings run freely and that bearings are not flooded by the forced oiling systems.

Glands

Glands are made in three styles. No style is equally well suited for all conditions, and the proper type is chosen to suit conditions of operation.

The Bronze Sleeve type requires only occasional lubrication from a grease cup provided.

The Labyrinth type is shown in principle in Fig. 3. It consists of a rotating sleeve, A, and a fixed casing, B. The many close contacts between the sharp edges of the teeth and the surfaces against which they fit keep leakage small. On back-pressure turbines what steam escapes is caught in leak-off space, E. It may be piped away if desired, but must have free escape or some steam may blow out of the end of the gland. A new labyrinth gland may require an hour or two of slow running to bring the many close contacts to a proper running fit.

The principle of a Carbon-Ring gland is shown in Fig. 4. The leak-off space serves the same purpose as in the labyrinth gland. In case of excessive leakage, look

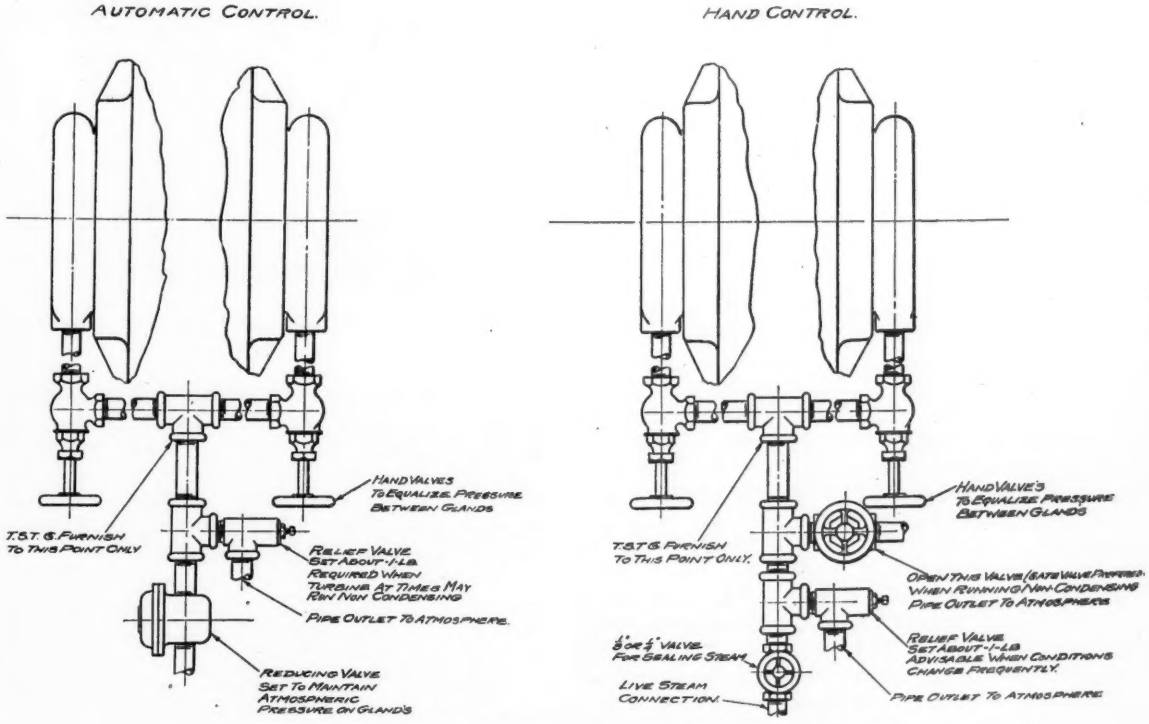

AUTOMATIC CONTROL.
HAND VALVES TO EQUALIZE PRESSURE BETWEEN GLANDS
T.S.T.C. FURNISH TO THIS POINT ONLY
RELIEF VALVE SET ABOUT 1-LB. REQUIRED WHEN TURBINE AT TIMES MAY RUN NON CONDENSING
PIPE OUTLET TO ATMOSPHERE.
REDUCING VALVE SET TO MAINTAIN ATMOSPHERIC PRESSURE ON GLAND'S
HAND CONTROL.
HAND VALVE'S TO EQUALIZE PRESSURE BETWEEN GLANDS
T.S.T.C. FURNISH TO THIS POINT ONLY.
OPEN THIS VALVE (GATE VALVE PREFERED) WHEN RUNNING NON-CONDENSING PIPE OUTLET TO ATMOSPHERE
1/8 OR 1/4" VALVE FOR SEALING STEAM
LIVE STEAM CONNECTION
RELIEF VALVE SET ABOUT 1-LB. ADVISABLE WHEN CONDITIONS CHANGE FREQUENTLY.
PIPE OUTLET TO ATMOSPHERE

FIG. 5. CONNECTIONS AND CONTROL FOR STEAM SEAL ON GLANDS

for a ring either broken or jammed into its slot by solids carried over with the steam. Rings may be refitted or replaced when necessary.

For condensing operation the leak-off space is connected as a steam seal and low-pressure steam supplied to prevent air from being drawn into the glands. The connections and arrangement for automatic or hand control are shown in Fig. 5.

Westinghouse Turbines

COMMENCING its turbine work under the American patents of Chas. A. Parsons, the Westinghouse Machine Co., has made many improvements and developed a variety of designs to meet special requirements in power plants and industries. While originally of the reaction type, later designs in the turbines of large capacity are provided with an impulse element for the first stage of the expansion. It is claimed that this element replaces without appreciable sacrifice of economy a considerable number of rows of reaction blading in the least efficient part of the reaction turbine, and makes possible a shorter and stiffer rotor. For the intermediate and low-pressure sections the reaction blading is still employed.

Of the modern high-pressure large-capacity turbines there are three general types, viz., the combined impulse and reaction single flow, the double flow and the semi-double flow turbines. These have many features in common, the main difference being in the arrangement of the blades on the drum.

Rotors for these turbines are built up of hollow steel shafts or drums which are machined both inside and outside. In the latest designs the low-pressure blading is carried on a solid disk made integral with the shaft and attached to the drum. The dummies are made up of separate steel rings.

The impulse blading is either machined or drop forged from special alloy metal. The blades are shrouded to prevent the steam from spilling over the ends, as will be noted in Fig. 1, which is a view of a portion of the semi-double flow rotor showing a section of a finished impulse blading and also a little of the reaction blading and the dummy ring. There are two rows of blades on the single-impulse element and the steam issuing from the first row is redirected onto the second row by a short section of stationary guide blades shown in Fig. 2. The reaction blades are slightly upset and a small hook or shoulder formed on one side, and the blade section near the base is increased, as shown in Fig. 3. The grooves in the rotor and cylinder are of dovetail section, and have a small auxiliary groove of rectangular cross-section in the bottom. The outer ends of the blades are tied together with the comma lashing. The blades are punched with comma-shaped holes, as shown in Fig. 3, and a wire of the same section is threaded through these holes. After the blades have been straightened and gaged, the part of the lashing wire corresponding to the tail of the comma is curled over, forming a rigid separator or distance piece between the blades. For the longest blades, two or more lashings are used.

The construction of the larger cylinders is a symmetrical design not likely to be affected by temperature strains. The intermediate blade rings are separate from the cylinder proper, being securely clamped into place. Whenever dummy packings are used they are made up in removable rings and in combination impulse and reaction turbines the nozzle chambers are also independent castings. In each turbine there are two nozzle chambers, the primary and the secondary; the secondary comes into action only when the turbine is loaded above its normal or rated capacity.

Packing the spindle against the inleakage of air is accomplished by means of a water gland. A bronze casting like the runner of a centrifugal pump is pressed on the shaft and rotates in an annular chamber surrounding the opening in the end of the cylinder through which the shaft projects. Water is fed to this chamber and under the action of centrifugal force a pressure is built up which effectively resists the inleakage of air.

Speed is controlled by the governor, Fig. 4, which is driven from the main shaft through a worm gear. The ball levers and links connecting these levers to the gov-

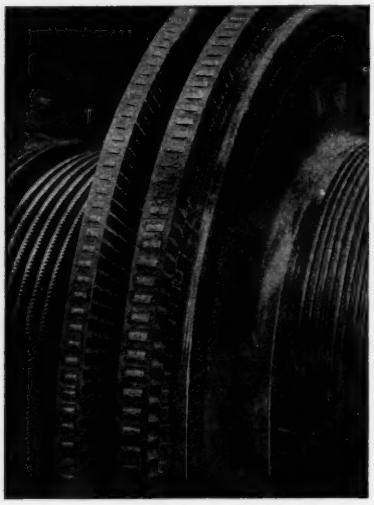


FIG. 1. PORTION OF A COMBINED IMPULSE AND REACTION ROTOR

ernor sleeve are all pivoted on knife edges. The main spring surrounds the spindle and bears directly on the governor sleeve. A small auxiliary spring with a manually or electrically operated tension gear for making small speed adjustments while running is connected to the governor linkage. It is used in synchronizing alternators or in distributing the electrical load among them.

In small turbines, the governor acts directly on the steam admission valves, opening first the primary valve, then, if necessary, the secondary, after the primary is fully open.

Governors for the larger turbines employ an oil relay mechanism for operating the steam valves. In these turbines, the lubricating oil pump maintains a higher pressure than is required for the lubricating system. The governor controls a small relay valve A, Fig. 5, which admits pressure oil to and exhausts it from the operating cylinder.

When the oil is admitted to the operating cylinder, lowering the piston, the lever C lifts the primary valve E. The lever D moves simultaneously with C; but on

account of the slotted connection with the stem of the secondary valve F, the latter does not begin to lift until the primary valve is raised to the point at which its effective opening ceases to be increased by further upward travel.

The operating valve A is connected not only to the governor, but also to a vibrator which gives it a slight

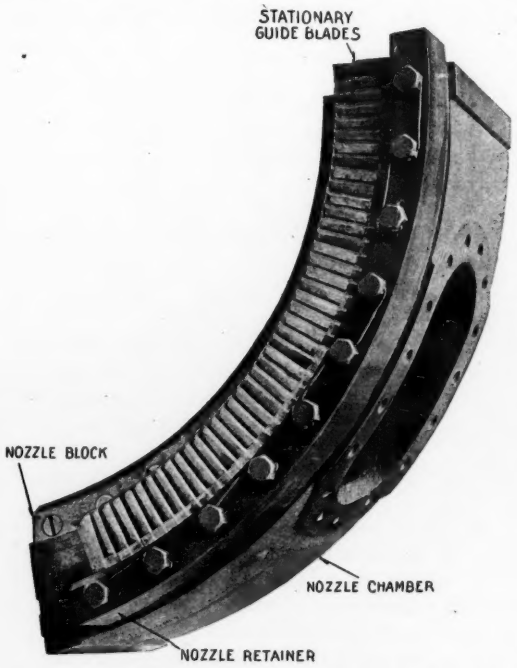


FIG. 2. NOZZLE CHAMBER WITH GUIDE BLADE SECTION

but continuous reciprocating motion while the governor controls its mean position. The effect of this is manifested in a slight pulsation throughout the entire relay system, which keeps it ready to respond instantly to the smallest change in the position of the governor.

Besides the main regulating governor these turbines are fitted with an entirely separate speed-limit governor

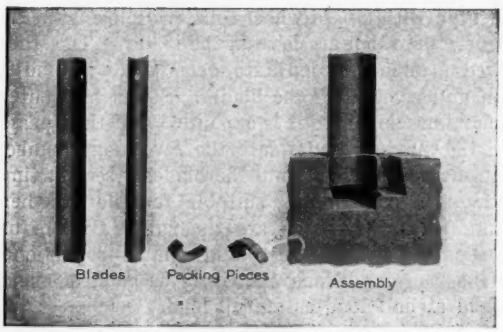


FIG. 3. REACTION BLADING

which acts automatically when the speed reaches the over-speed limit to trip a stop valve and shut off steam supply.

To compensate for an unbalanced condition within the limit of errors of the most refined methods, the bearings on high-speed turbines are made up of several concentric tubes with slight clearance between them. The lubricating oil fills these clearances and forms an elastic cushion and allows the spindle to find its true center of rotation. This nest of tubes is carried in a cast-iron sleeve which rests in a pedestal. On the outside of the supporting sleeve are four steel blocks fitted in slots spaced 90 deg. apart and secured with screws. The blocks extend above the outer circumference of the casting and are machined to form a section of a sphere. The

FIG. 4. TURBINE GOVERNOR

pedestal has a corresponding spherical bore, so that the combination forms a self-alining ball and socket bearing.

In the larger and slower running turbines, the critical speed is never reached and consequently the concentric oil-cushioned tubes are not required. The bearings for these turbines are similar to the supporting sleeve of the high-speed bearings, except that they are babbit lined

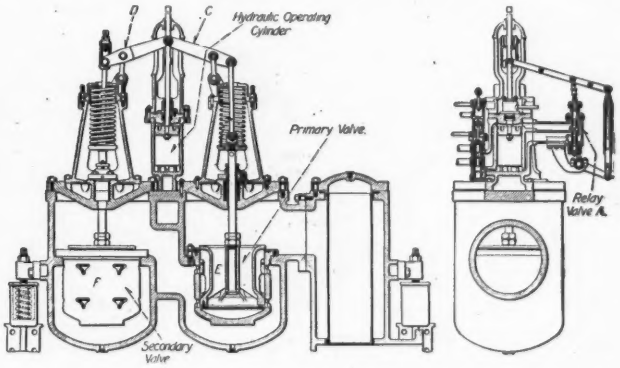


FIG. 5. VALVE GEAR WITH OIL RELAY

and made in halves. A large copper tube placed in a groove and covered over with the babbit lining of the bearing carries the lubricating oil from the pipe connection at the bottom to the bearing to the top from which point it is distributed over the bearing surface by means of grooves.

Lubrication is accomplished by a closed individual system which maintains a continuous circulation by means of a pump geared to the main shaft of the turbine. This system keeps the bearings flooded with cool oil at a moderate pressure. From the bearings the oil drains through a strainer to a cooler and is pumped back to the bearings.

In turbines in which an oil relay governing system is employed the oil pressure is maintained higher. The oil used for the valve mechanism passes direct to the relay and discharges to the cooler, while that used on the bearings goes through a reducing valve, thus maintaining only a moderate pressure of the bearings.

FIG. 6. HIGH SPEED TURBINE BEARING

Among the special turbines of large capacity the most important types made by the Westinghouse Co. are the automatic bleeder and the low-pressure turbines. The bleeder turbine automatically supplies the desired amount of low-pressure steam to the heating system, and at the same time efficiently develops the required amount of electrical energy and retains the operating characteristics peculiar to turbine apparatus. By referring to the sectional view, Fig. 7, the course of the steam through the turbine may be noted as follows: From the boiler steam is admitted through valve A, which is under the control of the governor G. After suitable expansion in nozzles N, it passes first through the impulse blading B, then the intermediate reaction blading B_1, thus converting into power the expensive energy of the steam. At the end of the blades B_1 the steam reaches an annular space S, from which it goes to the heating system through the outlet O. If the work done by the turbine is not sufficient to carry the power load, the valve A will automatically open wider and more steam will pass into the turbine, raising the pressure at S. At V is a weight loaded bypass valve which opens when the pressure is sufficiently great and allows the excess steam to pass into the low-pressure blading, B_2, where its expansion is completed before it passes to the condenser through the exhaust, X. By changing weights on the bypass valve, the pressure on the heating system may be maintained constant for any setting of the weights.

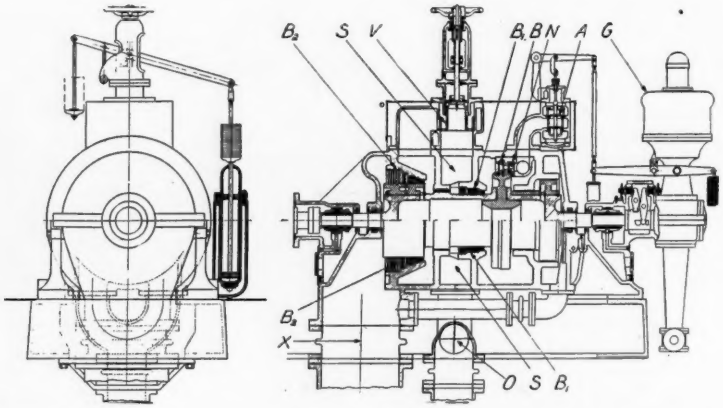


FIG. 7. SECTIONAL VIEW OF BLEEDER TYPE TURBINE

FIG. 8. VIEW OF ROTOR SHOWING NOZZLE AND REVERSER

The low-pressure turbine is of the double flow reaction type, with general features of construction similar to the high-pressure turbine with the high and intermediate sections removed. The features of special design have to do principally with the system of governing and supplying the steam which are individual problems worked out for each installation.

Westinghouse turbines of small capacity are of the impulse type with single wheel construction on the rim or periphery of which are located the blades. The wheel is grooved a sufficient depth to permit the individual insertion and riveting of the blades. Steam first enters the primary nozzle where it is expanded and then passes through the moving blades into the reversing chamber, where its flow is directed onto the blades a second time. This design is particularly suitable for plants where light and fluctuating loads are the rule. By the use of two separate nozzles, large and small, proportioned to suit the load conditions, relatively as good efficiency is obtained at half load as at full load.

These turbines run at high rotative speeds, but moderate blade velocities, and as usually installed employ a reduction gearing between the turbines and the driven machine.

GERMAN AIRPLANE ENGINES have cylinders made from steel forgings with sheet steel jackets welded in place.—The Gas Engine.

Auxiliary Apparatus

A BALANCED trip and throttle valve frequently used on turbine steam lines is shown in section in Fig. 1. When the pilot valve, which is carried by a pipe attached at P, is opened by the turbine governing mechanism, pressure under piston M is reduced while due to that above this piston, toggle link L is caused to snap over its center and the valve thereby tripped. The closing force is provided by the steam acting against the enlarged plunger on the valve stem.

Lever H may be employed to operate the valve by hand whenever so desired or required.

Figure 2 shows a typical automatic flow valve such as is employed in connection with mixed-flow turbines to prevent the existence of vacuum in the engine exhaust piping system, which would cause the drawing in of air through leaks and piston rod packings, thus impairing the efficiency of the condenser by overloading the air pump, and interfering with the automatic drainage of separators installed in the engine exhaust line.

Essentially this valve, which may be operated manually, if so desired, by a hand wheel provided for the purpose, consists of a housing in which are a number of spring loaded valves which remain raised from their seats as long as the pressure of the steam is above some predetermined value. With a drop of pressure below this predetermined value, however, the individual valves seat and thereby cut off further communication between the engine and the turbine.

Another valve generally termed a flow regulator, and frequently found on the low pressure supply lines of mixed flow turbines is illustrated in Fig. 3.

With the required low pressure existing, the enclosed piston below the valve proper is caused to descend, and with it the valve stem and attached disks. As the supply diminishes, however, the pressure above the piston decreases and due to the action of the counterweight, the moving member is caused to rise and the steam flow thereby discontinued. A dash pot with adjustable bypass insures a steady movement of the parts.

Frequently extraction lines connected to one of the stages of a turbine are supplemented with steam from other sources, and as a consequence these lines must be equipped with some form of non-return valve to prevent a reverse flow of steam into the turbine. A valve for such a purpose is shown in Fig. 4. A dashpot provided insures noiseless operation without pounding of the seat, and where the weight of disk and spindle is considerable, a spring in the hollow bottom port counteracts this excess weight.

Installed in the Vandergrift Works of the American Sheet & Tin Plate Co. is a regenerator supplying exhaust steam to a Curtis mixed-pressure turbine which, however, does not at times take all of the steam available. The surplus is permitted to pass to a Rateau low-pressure turbine by means of the duplex shutoff valve shown in Fig. 5, which simultaneously shuts off the live steam supplied to this turbine at other times. This valve, which was manufactured by the G. M. Davis Regulator Co., is electrically operated by means of a solenoid.

Oiling Systems

In the smaller sizes of steam turbines, some form of ring oiler is generally employed, but in machines having heavy rotors operating at high rates of speed, more positive means of lubrication must be provided. This, in the majority of cases is accomplished by equipping the turbines with individual systems, the oil being delivered to the bearings under pressure by pumps operated either direct from the rotor shaft or by an independent steam cylinder. Frequently both of these types are used.

All parts of the De Laval turbine requiring lubrication are supplied with oil by a circulating pump of the positive gear type, driven from the lower end of the governor spindle. The supply of oil is drawn through

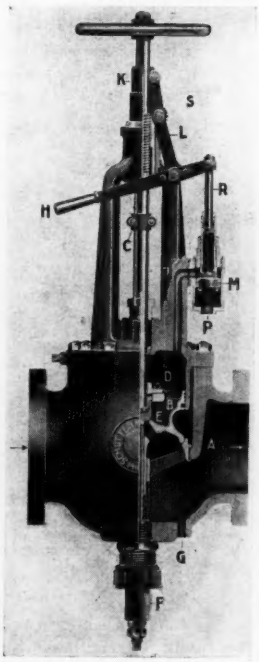


FIG. 1. SCHOTTE BALANCED TORBO THROTTLE VALVE

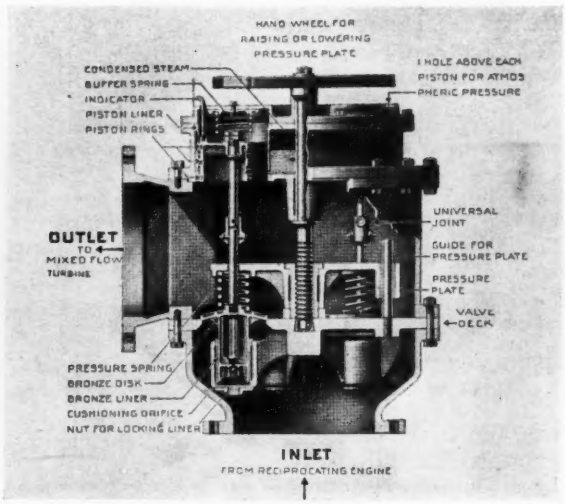


FIG. 2. COCHRANE MULTIPORT AUTOMATIC FLOW VALVE

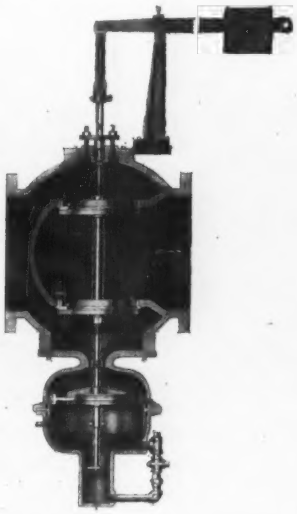


FIG. 3. DAVIS FLOW REGULATOR

screens or filters from a reservoir built integral with the bed plate of the machine, and is forced to an elevated storage tank, whence it flows by gravity to the sight-feed lubricators on each bearing.

Allis-Chalmers turbines are equipped with two pumps, one a steam operated unit for use during starting periods, and the other, of rotary type which may be omitted if the plant is provided with a central oiling system, the oil under such conditions being taken direct from the station supply tank.

After having been passed through the bearings, and before being re-used, the oil is caused to flow through a nest of tubes surrounded by water, wherein it is cooled. A portion of the boiler feed supply may be used for this purpose, thus conserving heat which otherwise would be wasted.

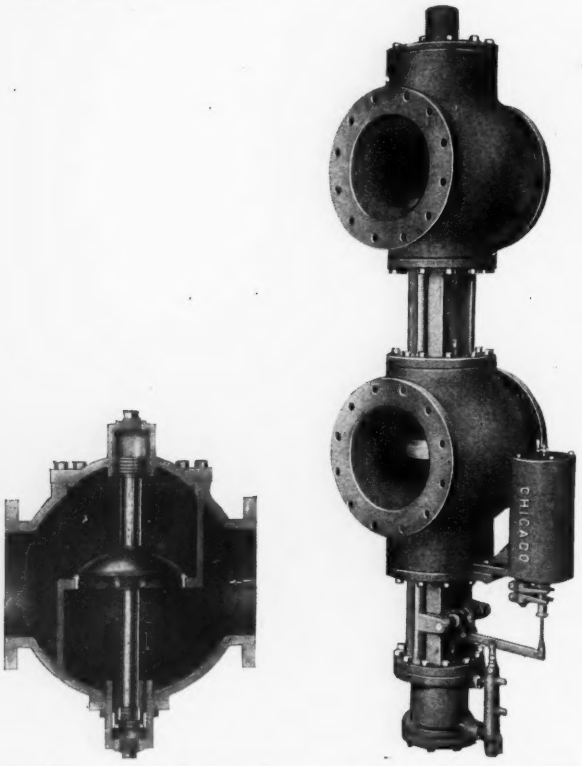


FIG. 4. DAVIS TURBINE BLEEDER LINE NON-RETURN VALVE
FIG. 5. DAVIS DUPLEX TURBINE SHUT-OFF VALVE

Oil supplied to the bearings of Curtis turbines is taken from a general distributing system. Normally the pressure in this is maintained by a simple geared oil pump, driven from the main turbine shaft; in large machines an independent steam driven pump is also supplied.

Where bearing pressures and speeds so require, cooling of the oil is effected by water circulating through coils of copper pipe embedded in the babbit as shown in Fig. 6. Inlet and outlet connections of these coils are made from the outside of the main bearing pedestals and each branch is provided with a plug valve for adjusting the water supply. The discharge water passes through a sight discharge coupling into the drain pipe.

In the Westinghouse turbine, a closed system is employed through which a continuous circulation is maintained by means of a pump geared to the main shaft, which keeps the turbine and generator bearings flooded with oil at a very moderate pressure. From the bearings the oil drains through a strainer into a collecting reservoir; thence it is pumped through a water cooler and back to the bearings for re-use.

In machines in which the oil relay governing system is employed, and a high pressure has to be maintained by the pumps, the comparatively small quantity of oil required for operating the valve mechanism passes to the relay cylinder, whence it exhausts into the cooler. The remaining circulating oil discharges directly into the cooler through a spring-loaded pressure reducing valve so that the pressure in the main system is the same whether the oil relay governing system is employed or not.

Drains

In all turbine installations a drain pipe should be provided outside of the throttle valve to carry off any condensation which may tend to form and collect in the steam line. This should connect with a trap provided for the purpose.

Drains from the various stages of multistage machines may be connected to a main drain line which may communicate with a trap, or preferably may be connected to the exhaust chamber. Each of these drains should be fitted with a globe valve which should be opened when starting the set, or whenever vibration occurs due to the accumulation of water in the lower half of the casing. In turbines such as the Curtis, where some of the nozzles between stages are located

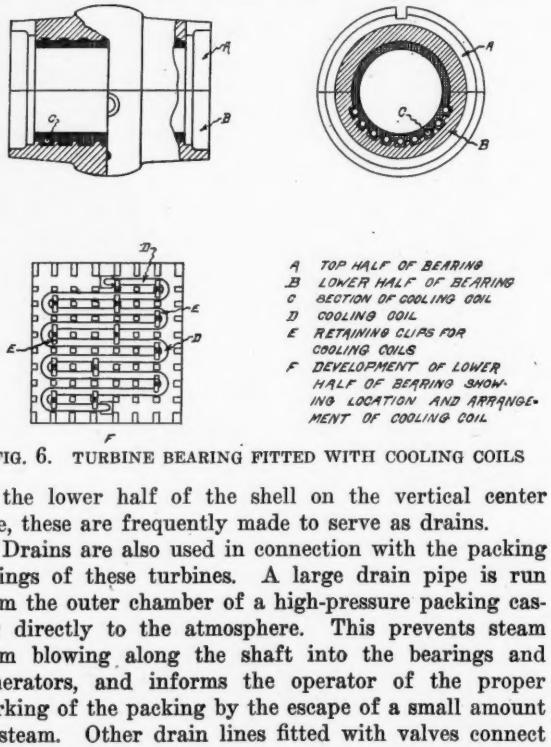


FIG. 6. TURBINE BEARING FITTED WITH COOLING COILS

in the lower half of the shell on the vertical center line, these are frequently made to serve as drains.

Drains are also used in connection with the packing casings of these turbines. A large drain pipe is run from the outer chamber of a high-pressure packing casing directly to the atmosphere. This prevents steam from blowing along the shaft into the bearings and generators, and informs the operator of the proper working of the packing by the escape of a small amount of steam. Other drain lines fitted with valves connect with the inner chambers and the lower pressure stages.

A drain should also be provided at the lowest point of the atmospheric exhaust line and where this is of such construction as to employ blank flanges, it is

advisable to connect an additional drain to the lowest point of each blanking plate.

In a Parsons steam turbine, with the blades secured in the cylinder by means of separate calking strips inserted between the blades, the condensation in the cylinder can drain away through the blades to the exhaust of the turbine and so to the condenser. In the Allis-Chalmers steam turbine, where the blades are secured in foundation rings which project above the surface of the condenser, drainage is achieved by cutting suitable small grooves through the foundation rings, the bottom of the groove flush with the bore of the cylinder. These grooves are sufficient to drain away any condensate which may collect between the rows of blades, but are so small that steam leakage is negligible.

Turbine Vibration

Causes of and Remedies for Many Turbine Troubles Which Develop During Operation

ONE of the most prevalent causes of turbine vibration is incorrect alinement. This may result in vibration only at heaviest loads giving a slow variable periodicity. A turbine may run perfectly for a time after starting up, but after a while start vibrating which lasts until it is shut down. Checking for alinement while the turbine is hot will usually reveal the cause of trouble. The difference in temperature of a machine running condensing and non-condensig may cause vibration under one condition, yet give perfect operation under the other. Flexible steel foundations are a frequent cause of poor alinement and consequent vibration of the turbine under operation. Self-alining bearings should not be clamped too tightly, as this cause sometimes gives intermittent vibration. Out-of-line troubles are sometimes the results of pipe strains due to temperature changes without expansion provisions.

Vibration Caused by Unbalance

Although turbines are carefully balanced in the factory, this balance may be disturbed during shipment, in which case only careful readjusting will put the machine in perfect operating condition. This cause is characterized by an even, continuous shake at no load, slightly better at full load, with the same periodicity as the speed. Unbalance vibration may be caused by the action of acid in steam upon the balancing weights or dirt upon the buckets, or even unbalanced coupling castings.

Bearings and Lubrication Troubles Cause Vibration

Oil for turbine bearings should stand a higher temperature than in engine practice and should not saponify or foam when coming in contact with water. One of the first effects of oil failure due to insufficient supply, breaking down of film or poor quality is violent vibration accompanied by heating at the bearings.

Bearings which have been incorrectly assembled may cause the oil passages to be stopped up, thus causing vibration. Bearings which are too tight or too loose may not necessarily cause vibration if other adjustments are perfect, but tend to magnify vibration due to other causes. Bearings are made 0.0005 to 0.003 in. per inch of shaft diameter larger than the shaft depending upon the condition required.

Vibration Due to Packing Troubles

Packing rings so small in diameter as to pinch the shaft will cause violent vibration accompanied by heating of the shaft. Jamming action often occurs with machines having metallic packing rings when first started up; the remedy is to turn the machine over slowly until the packing is worn to the shaft.

In fitting packing rings, those on the outside, which separate the atmosphere from a small pressure may be fitted so as to hug the shaft and usually are so designed. Other rings, subjected to greater pressure, should be given some play amounting to 0.001 to 0.002 in. per inch of shaft diameter, which will prevent binding and consequent vibration.

Foundation Troubles

Foundations should be homogeneous throughout for supporting the entire unit as variation in temperature or uneven settling may cause poor alinement and vibration. Sympathetic vibration is often neglected by engineers in laying out foundations and steel structures are often provided that need considerable bracing before being fit for a turbine. Improper grouting is sometimes a cause of vibration, as it causes the alinement to become out of adjustment.

Water in Steam

Small, stout turbines of one or two stages are not affected by water coming over with the steam beyond slowing down the speed or producing a small vibration of low periodicity; but in large turbines it sometimes produces vibration so serious as to necessitate shutting the machine down. The vibration resembles very closely that due to unbalance.

Changing from condensing to non-condensing, or vice versa, causes contraction or expansion of the rotor, sometimes resulting in rubbing between moving and stationary blades. The machine should be operated under the conditions for which it was designed to avoid troubles from this cause.

Vibration Due to Mechanical Construction

Turbines are notably free from vibration trouble caused by poor mechanical construction which is probably due to the extreme care taken in manufacturing and testing the machines. Among the things for which to be on the guard when operating turbines are the following: Loose wheels on a shaft do not always cause vibration, but are active in magnifying it. Vibration from this cause is likely to be of a variable character, and on shutting down or starting, noises may be heard in the casing.

Loosely constructed built-up wheels readily magnify vibration and the effect will seldom remain unnoticed.

Vibration is sometimes caused by a flat spot on the shaft or by an eccentric coupling; this is characterized by its regularity, generally at low speeds and often by knocking.

Defects in minor shafting gears, etc., are usually characterized by the locality in which they appear.

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Acknowledgement

In preparing a resume of current practice and knowledge on any technical subject there is little chance for originality, except in the method of presenting the subject. The information is gleaned from authoritative books, from manufacturers and from men experienced in the profession. Under such conditions, it is hopeless to try to acknowledge all help that has been given or sources consulted, and we must be content with expressing our thanks to all who have assisted in the work of preparing this issue, in which the facts of modern steam turbine practice are presented. They, like ourselves, will be repaid by the help and satisfaction that readers get from the use of the issue.

A Look Backward

The year 1917 has been filled with great events for this country. In the first months, we feared that complications would not permit us to keep out of war, but hoped that some way might be found. We did not then comprehend fully the aims of the German government, but we early found them out. War came, and we have made some progress toward preparing for it. Perhaps the longest step we have taken is in the consolidation of public opinion as to what must be done—that the war must be won at any cost.

Material progress in preparation has been cheering in some directions, disheartening in others. We have built cantonments for training our army and navy; we have assembled great numbers of men for the training, and have sent abroad many who are ready for service. Much has been done, also, towards organizing our governmental machinery and our industries for the conduct of war; great success has followed the drives for support of the Red Cross, the Y. M. C. A., Knights of Columbus and other auxiliary agencies for the care of our soldiers; and Liberty Loans have been well supported.

But in many other things we have fallen short of our goal, and of our possibilities. While shipbuilding, which is vital to our success, has been much talked about, there has been much delay in getting started, and some vacillation in policy. There is still disagreement as to whether main reliance shall be placed on wooden ships or on steel, with the probability that the decision must rest on ability to get materials. Lumber has been so much in demand for cantonments that the mills could not reserve the necessary supply for the shipyards, but a great campaign is now under way to speed up production and insure that no lumber needed for ships shall be diverted to other uses, and it is predicted by Mr. J. E. Rhodes of the Southern Pine Association that by next summer we shall catch up with our schedule, and be able to operate the shipyards 24 hr. a day.

In this, as in other matters, one limiting factor is transportation. Railroads have strained at every point to make best use of existing facilities by co-operation between roads, cutting down delay in loading and unloading, and loading to over capacity. But the increase in volume of business has more than offset the gains, and cars are lacking to handle shipments promptly. A serious consequence of this has been the cutting down of coal production. Bituminous miners work only when

cars are available to ship the coal; during the weeks ending Nov. 24 and Dec. 1 mines reported the loss of production due to car shortage as averaging 20 per cent, running in some large Pennsylvania and West Virginia regions to as high as 40 per cent. In the present scarcity of coal, this condition is serious.

Recent investigations indicate that those departments specially charged with supplying equipment for the soldiers are just now getting under sufficient way to give promise of adequate results. While we are suffering from a policy of non-preparedness, adhered to after danger of war was evident and up to the time that war was declared, speed since the declaration of war does not seem to have been considered of primary importance. And one disappointing feature is that some of our officials, even after this experience, cannot see the vital need of taking precautions that the nation shall never again be caught in such a condition.

While the great majority of the people are giving every aid to help in our preparation for the struggle, some elements have not yet realized that their present personal profit will be of little use if we lose. They are giving all their attention to securing the most possible for themselves out of the present needs of the government for men and products, forgetting or ignoring their duty to the land which affords them their opportunities. Profiteering, by either manufacturers or workmen, is disloyalty, and much of our failure to make the progress desired in national preparation must be laid to just this kind of citizen.

On the whole, those things that have to do with national spirit to win and with man-power for the fighting forces may be said to have progressed well. Equipment and the organization of man-power for industrial needs have lagged behind.

A Look Forward

What of the coming year? That it must be one of sacrifice and endeavor is certain; but no wise man will attempt to predict its happenings. Peace we hope and pray for, but not until those things are secured for which we entered the fight.

To this end, every man has certain duties. Economy in food, fuel and clothing; support of government loans and purchase of war savings certificates; the cheering of soldiers and sailors to keep up their spirits; effort to speed up production; refraining from purchase of non-essentials which take labor away from essential industry.

Surely no real man wants to eat more than he needs or indulge an appetite for meats or sweets and feel that he is thereby causing some soldier or refugee in France to go hungry. Just as surely, we should not wish to buy new clothing while the old will serve, and thereby bring suffering to one of our men in the trenches.

War savings certificates and thrift stamps bring the possibility of aiding to finance the war within the reach of every person, as well as offering a good investment, so that selfish interest as well as loyalty plead for this activity.

But what about the rest? The officers tell us that letters to the men in camp and across the ocean are one of the greatest helps in cheering them up and breaking the monotony of army life. Are we all writing to some fellow to assure him of our interest and support? A half-hour once a week isn't much to give when he is giving all his time and strength to fight for our country.

And a fundamental need is for more productive labor. The spectacle of workmen stopping the progress of shipbuilding and of coal production has appalled the men from England. We must have fuel, ships, transportation, arms, ammunitions, artillery, shells, clothing, food—or our building of cantonments and training of men is a waste of effort. Workmen of England have realized that they at home must sacrifice as well as those who are at the front, if the army is to succeed. All restrictions on hours of labor and amount of output have been given up for the period of the war, and agreement made that differences as to wages shall be settled by arbitration without stopping production by resorting to strikes. We must recognize the necessity of accomplishing the same increase of production and freedom from loss, although we may not use the same means to accomplish it.

Employers must realize, too, that they have a duty of sacrifice; that profits are of no importance compared with duty to the country, and that the main thing is so conducting their business that the workers may be assisted in turning out the greatest amount possible of those things needed for carrying on the war. Individual interests must stand aside when the needs of the nation are pressing, and if this does not come about voluntarily, the government may be forced to appoint a man, or a board, to do in the matter of labor and employment what the Food Administration is doing in its field. Selfish struggle for personal advantage cannot be allowed to interfere with winning the war.

Manufacture of non-essentials will be curtailed largely by decrease in demand from buyers, but the machinery and organization of the factories can in considerable degree be kept busy by changing their output to meet government needs. Some industries will probably have to cease, and the workers find employment in other lines, as has been the case in France and England. But, as a British leader has said, the man who cannot change his mind or his occupation over night is hopelessly behind in present conditions. It is to be hoped that some definite announcement may be made of what industries or classes of industries are to be considered non-essential in controlling supply of materials in time, so that gradual readjustment may be made, but so far the administration has not been willing to put itself on record.

One thing is certain. The United States and its people must learn more of self-denial and self-sacrifice during the coming year than they have yet learned. We have made a beginning, but it is not enough. We are achieving the will to do and suffer, but we do not yet realize the height of achievement and the depth of suffering to which we must attain before we can win. And to secure a willing endurance of that sacrifice and suffering, we must find means by which the load will be equably divided, so that each shall bear a burden according to his strength.

But all must share in carrying the burden. There must be no slackers and no obstructionists.

May each of us feel sure at the end of the year that he has done his full part as a man and a loyal citizen of these United States of America.

A New Year Thought

IT'S the kindly hearts of earth that make
This good old world worth while,
It's the lips with tender words that make
The care-erasing smile.
And I ask my soul this question when
My goodly gifts I see—
Am I a friend to as many men
As have been friends to me?

When brothers speak a word of praise
My wavering will to aid,
I ask if ever their long, long ways
My words have brighter made.
And to my heart I bring again
This eager, earnest plea—
Make me a friend to as many men
As are good, staunch friends to me.

E. S. A.

Catalog Notes

NEW ROCHESTER clutch drive automatic lubricators are illustrated in a booklet from Greene, Tweed & Co., 109 Duane St., New York.

LINK-BELT Book No. 310, entitled "The Ideal Drive for Clayworking Machinery," and illustrating the use of Link-Belt Silent Chain, has just been received from Link-Belt Co., 39th St. and Stewart Ave., Chicago.

AUTOMATIC FLUE BLOWERS are the subject of a folder giving the advantages and directions for installing those made by A. S. Dillon Automatic Flue Blower Co., 2443 Olive St., Kansas City, Mo.

ANNOUNCEMENT IS MADE that the 1918 edition of "Motor Trucks of America," of which S. V. Norton, manager of truck tire sales of the B. F. Goodrich Rubber Co., is the author, will be ready for distribution on Jan. 1. The book, consisting of 200 pages of reading matter, illustrations and specifications, is the sixth to be issued in as many years. It is hand sewn and of heavy binding.

Complete specifications of nearly 150 of the leading gas and electric-driven motor trucks are contained in the volume. In fact, there are 31 specifications for each model. These details are absolutely dependable as the approval of each truck manufacturer is obtained before specifications of his model are inserted. For the reason that it offers opportunity to the buyer and manufacturer to compare models, lists all makes according to sizes and affords a perspective on changes of design by comparison with previous editions, "Motor Trucks of America" is accepted throughout the motor world as the one complete and authentic handbook of the truck industry.

The illustrations, too, add much value to the book. They picture the very latest models and furnish the man in the market for a truck, or the manufacturer themselves, for that matter, suggestions for special bodies for particular line of work.

A NEW CIRCULAR from Homestead Valve Mfg. Co., Homestead, Pa., illustrates a few of the many types of Homestead locomotive blowoff cocks.

POWER PLANT ENGINEERING

Trade-mark, Reg. U. S. Pat. Off.

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Circulation of this issue, 22,750.

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537 S. Dearborn Street

Chicago

Position Wanted

POSITION WANTED—By engineer experienced with producer gas and oil engines, a.c. and d.c. equipment, pumps, storage batteries, interior wiring and repair work. Technical graduate. References. Address Box 501, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. 1-1-1

POSITION WANTED—By young man as oiler in large central station, with 7 years' experience with Corliss engines up to 400 hp., d.c. generator. I. C. S. graduate on complete Steam Engineering. Age 27. Married. Best of reference. Will go any place. Address Box 493, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. 11-15-2

POSITION WANTED—As helper or oiler in small or large plant. Will come at once. Sober and industrious. Will consider reasonable salary. Address Box 494, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. 10-1-4

POSITION WANTED—By young man as oiler or assistant engineer in or around Chicago, with chance of advancement. Eight years' experince; ready to start work at a week's notice; references. Address Box 492, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. 11-1-2

POSITION WANTED—By steamfitter with 17 years' experience at the trade. Can furnish references as to habits and ability to handle men. I am 33 years old. Married. Address J. H. D., 669 Wunderlick St., Barberton, O. 10-15-2

POSITION WANTED—By practical engineer of 15 years' experience. Am familiar with a.c. and d.c. generators and motors, also refrigerating machinery. Am 38 years old, married and strictly sober. Good references. E. W. Jones, Genl. Del., Metropolis, Ill. 11-15-2

POSITION WANTED—As engineer or oiler in power or ice plant. Have had 15 years' experience. Fred J. Waite, 659 Douglas Ave., Aurora, Ill. 10-15-2

POSITION WANTED—Chief engineer desires change of location. Competent to handle large plant. Experienced with a.c. and d.c. machinery, irrigating machinery, ammonia compressors (practical ice-maker), gas engines and air compressors. Can install and o repairs. Not liable to conscription. Address Box 495, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. 10-15-2

POSITION WANTED—As chief engineer. Am familiar with a.c. and d.c. generators and refrigerating and steam heating plants. I desire change. I can furnish good references and have been with present employer about 4 years. Age 35. Single. Address Box 715, Middleton, Conn. 12-1-1

POSITION WANTED—As chief engineer for manufacturing plant by technical graduate. Four years as chief in present plant. Age 30. Married. A1 references. Present salary $1800. R. R. Rogers, 32 Cordova St., St. Augustine, Fla. 1-1-1

POSITION WANTED—As watch engineer in power or ice plant. Ten years' experience. Married and sober. Frank Wells, 210 Pearl St., Jeffersonville, Ind. 1-1-1

Help Wanted

WANTED—Resident subscription agents in the following cities: Atlanta, Ga.; Cincinnati and Columbus, Ohio; Kansas City, Mo.; Los Angeles, Cal.; Louisville, Ky.; Newark, N. J.; New Orleans, La.; Portland, Ore.; St. Louis, Mo.; San Francisco, Cal.; Washington, D. C. This is an excellent chance for a hustler, who is familiar with power plant conditions, to turn his spare time into dollars. To such a one we offer a Real Opportunity. Write Subscriptin Department, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. tf.

MANAGER WANTED—Steady young man with business, electrical and steam engine experience, to manage electric light and ice plant in town of 3,000. Salary $90 to start. Living cheap. Give education, experience and references. Man from Middle West preferred. This plant is owned by a syndicate with good opportunities for advancement. Missouri Valley Light & Power Co., El Dorado Springs, Mo. 12-15-2

BOILER ROOM FOREMAN WANTED—For 5000 b.hp. plant, Stirling and B. & W. boilers, induced and natural draft, economizers, chain grate stokers. Must be able to handle operation, maintenance and efficiency work and get results, None but thoroughly competent and experienced men need apply. Middle West city of 60,000. State age, experience, etc., fully and present salary. Address Box 503, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. 1-1-2

Wanted

WANTED—Agents on commission to sell our shaking grates. We will install grates on trial or put it to test at our expense with any grate made. Address Armstrong Mfg. Co., Springfield, Ohio. tf.

WANTED—Agents handling engine and boiler room specialties in every section for side line paying large commissions. Give experience, line carried and territory covered. Box 381, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill.

WANTED—Marine boilers. We will pay cash awards for information that leads to a purchase by us of marine boilers of from 75 hp. to 250 hp. The Pittsburg Boiler & Machine Co., Pittsburg, Kan. tf.

WANTED—Agents who call upon engineers to sell positively the best pump valve on the market Commission basis. Excellent proposition. Address Box 500, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. tf.

WANTED—Good second-hand steam engine indicator outfit, with carrying case. Give description and lowest price in first letter. Address Box 502, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill. 1-1-1

WANTED—Agents handling engine and boiler room specialties in every section for our Boiler Girth Seam Protector. Liberal inducements. Give experience and territory covered. Address National Boiler Protector Co., Dayton, Ohio. tf.

For Sale

FOR SALE—Atlas Engine, ready for prompt shipment and immediate use. Size 22x30, left-hand drive wheel, 12 ft. in diameter, 3-ft. face, shaft 10 in. in diameter with wheel governor. This engine was developing 500 hp. when taken from our plant and is in very good running condition. Will sell for $800. Address The Parker-Young Company, Lisbon, New Hampshire. 1-1-2

FOR SALE—Gas engine, 25 hp., run 11 months. A. Birch & Bros., Somerville, Mass. 12-1-3

BOILERS—Two 300 hp. Babcock & Wilcox W. T. boilers with stokers. Insurance certificate Dec. 1, 1917 for 160 lbs., $7000. Two 500-hp. Heine with stokers. Insurance certificate Dec. 1, 1917 for 160 lbs, $12,000. Power Machinery Exchange, Inc., 1 Montgomery St., Jersey City, N. J. tf.

Patent and Patent Attorneys

PATENTS THAT PROTECT AND PAY—Advice and books free. High references. Best results. Promptness assured. Send sketch or model for search. WATSON E. COLEMAN, Patent Lawyer, 624 F St. N. W., Washington, D. C. tf.

PATENTS—C. L. Parker, Attorney-at-Law and Solicitor of Patents. Patents secured promptly and with special regard to the legal protection of the invention. Handbook for inventors sent upon request. Washington, D. C. tf.

PATENTS—H. W. T. Jenner, Patent Attorney and Mechanical Expert, 606 F St., Washington, D. C. Established 1883. I make a free examination and report if a patent can be had and the exact costs. Send for full information. tf.

A. P. CONNOR, Consulting Electrical and Mechanical Engineer, Attorney-at-Law and Solicitor of patents and Trade-Marks. Results guaranteed. 121 Carroll St., S. E., Washington, D. C. tf.

PATENTABLE IDEAS WANTED—Send for three free books. R. B. Owen, 28 Owen Bldg., Washington, D. C. tf.

Technical Books

FREE ENGINEER BOOKS and power plant tools and specialties, for securing new subscribers for Power Plant Engineering. A dandy chance for the engineer with a little spare time to get some fine premiums. Send for free descriptive circular and terms. Subscription Department, Power Plant Engineering, 537 S. Dearborn St., Chicago, Ill.

MISCELLANEOUS

Miscellaneous

FACTORY MANAGERS, ENGINEERS AND OWNERS—Coal is high and scarce. Let us tell you how to conserve it and utilize your exhaust steam for heating and drying purposes. Address Monash Engineering Co., 1413 W. Jackson Boulevard, Chicago, Ill. tf.

Sparks From The Advertising Pages

Where Buyers and Sellers Meet

E Pluribus Unum

Strictly speaking, "welding" is the joining of two pieces of metal by hammering them together while they are hot enough to be plastic. In this sense "welding" is limited to the blacksmith shop.

But custom has enlarged the meaning of the term to take in not only the joining of metals by chemical and electrical methods, but the welding of human efforts to form a strong and binding whole.

E Pluribus Unum—"In Union there is strength" is the way our founders tried to weld the colonies into a nation by means of the constitution.

The trait of *unified* endeaver is distinctly American—team-work is our national specialty. The result of this principle of cooperative action always justifies its use.

The Chief Engineer of the power plant does not attempt to stoke his boilers, oil his machinery and perform all of the many necessary duties of the plant. He depends upon the *welded* efforts of competent assistants for the efficient running of his machinery.

So in matters of policy and in the adoption of new efficiency methods, the Chief Engineer naturally depends upon his fellow engineers for cooperation in arriving at the best results.

The work of many capable minds, united in purpose, is better than the individual effort of the most brilliant mind among them. In the every day working of the plant, in deciding new issues as they come up, and in buying new equipment, the Chief Engineer needs the *welding together* of all the talents he can command.

This is especially true of buying—for he does not always have at his elbow someone with unbiased judgment to help him decide which machinery is the best to install—which will give the best account of itself on the job.

POWER PLANT ENGINEERING is a friend whose judgment deserves consideration—because it is sincere and because its advertising pages are a directory of the country's best power plant equipment.

Let friend POWER PLANT ENGINEERING cooperate with you always, for you can safely use its advertising columns as your buying guide.

Welding of your ability to judge your needs with the service best equipped to meet those needs is bound to follow habitual study of these advertisements.